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Ochiai et al.

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[54]	CVT RATIO RATE CONTROL AS A FUNCTION OF THROTTLE RELEASE RATE	4,597,308	7/1986	Tanaka et al.	477/46
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[75]	Inventors: Tatsuo Ochiai, Fujisawa; Hirofumi Okahara, Isehara, both of Japan	4,817,469	4/1989	Shigematsu et al.	477/46 X
		4,831,898	5/1989	Miyawaki	477/46 X
		4,853,858	8/1989	Kumura	477/44 X
[73]	Assignee: Nissan Motor Co., Ltd., Yokohama, Japan	5,047,937	9/1991	Vahabzadeh et al.	364/424.1
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[21] Appl. No.: **430,724**

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[22] Filed: **Apr. 28, 1995**

405215211 8/1993 Japan 477/46

[30] Foreign Application Priority Data

Apr. 28, 1994 [JP] Japan 6-092018

Primary Examiner—Khoi Q. Ta
Attorney, Agent, or Firm—Foley & Lardner

- [51] Int. Cl.⁶ **F16H 9/00; F16H 59/24; F16H 59/36; F16H 61/18**
- [52] U.S. Cl. **477/46; 477/48; 477/905**
- [58] Field of Search **477/44, 46, 48, 477/49, 905; 364/424.1**

[57] ABSTRACT

A control system for an engine-CVT drive-train includes a ratio control element positionable to various command positions in response to a rate of change per unit time of CVT ratio determined by a controller to establish various CVT ratios. The controller decreases the rate of change per unit time of CVT ratio as the rate of change per unit time of the position of a gas pedal toward a release position thereof increases.

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2 Claims, 13 Drawing Sheets

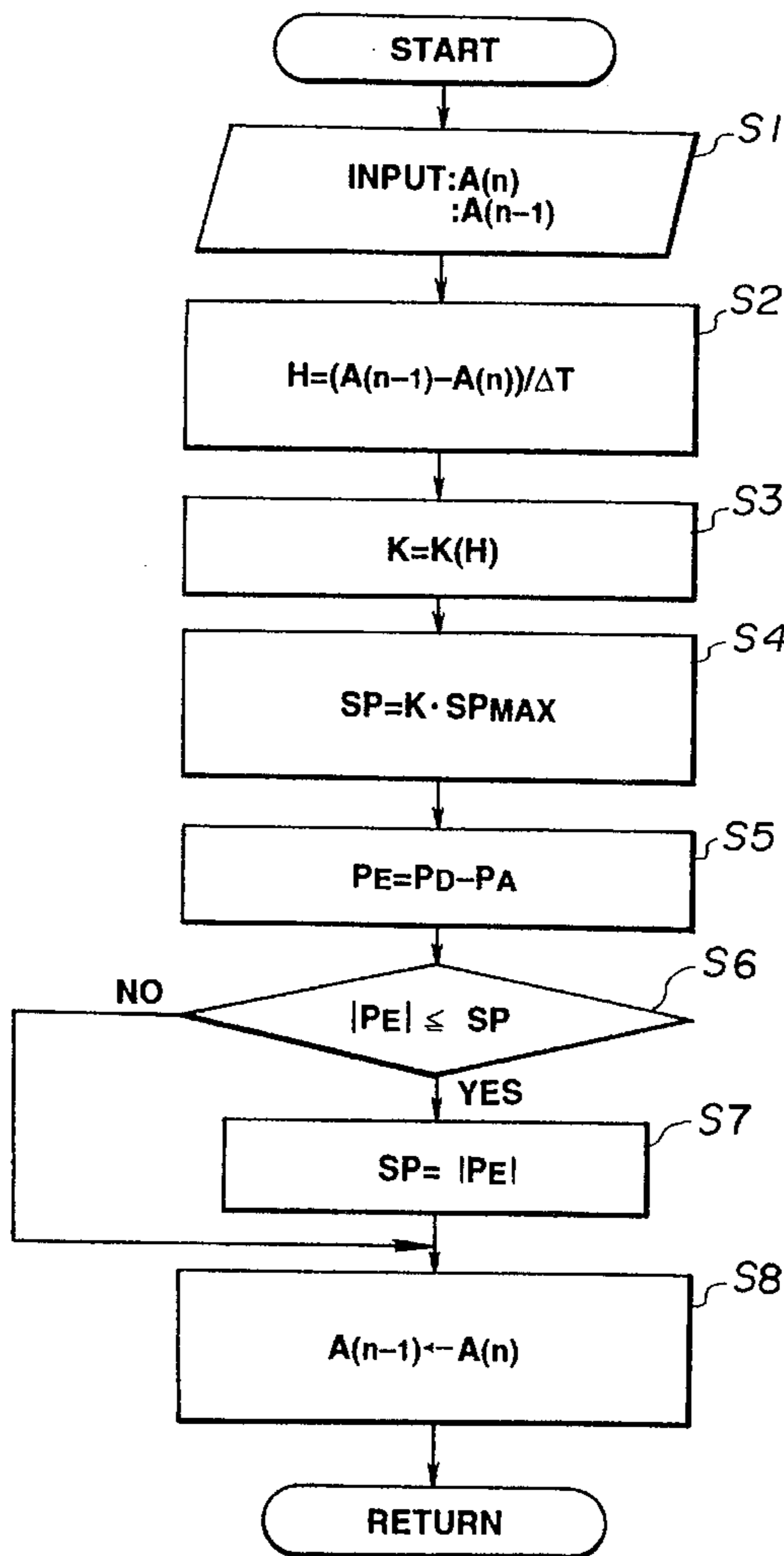
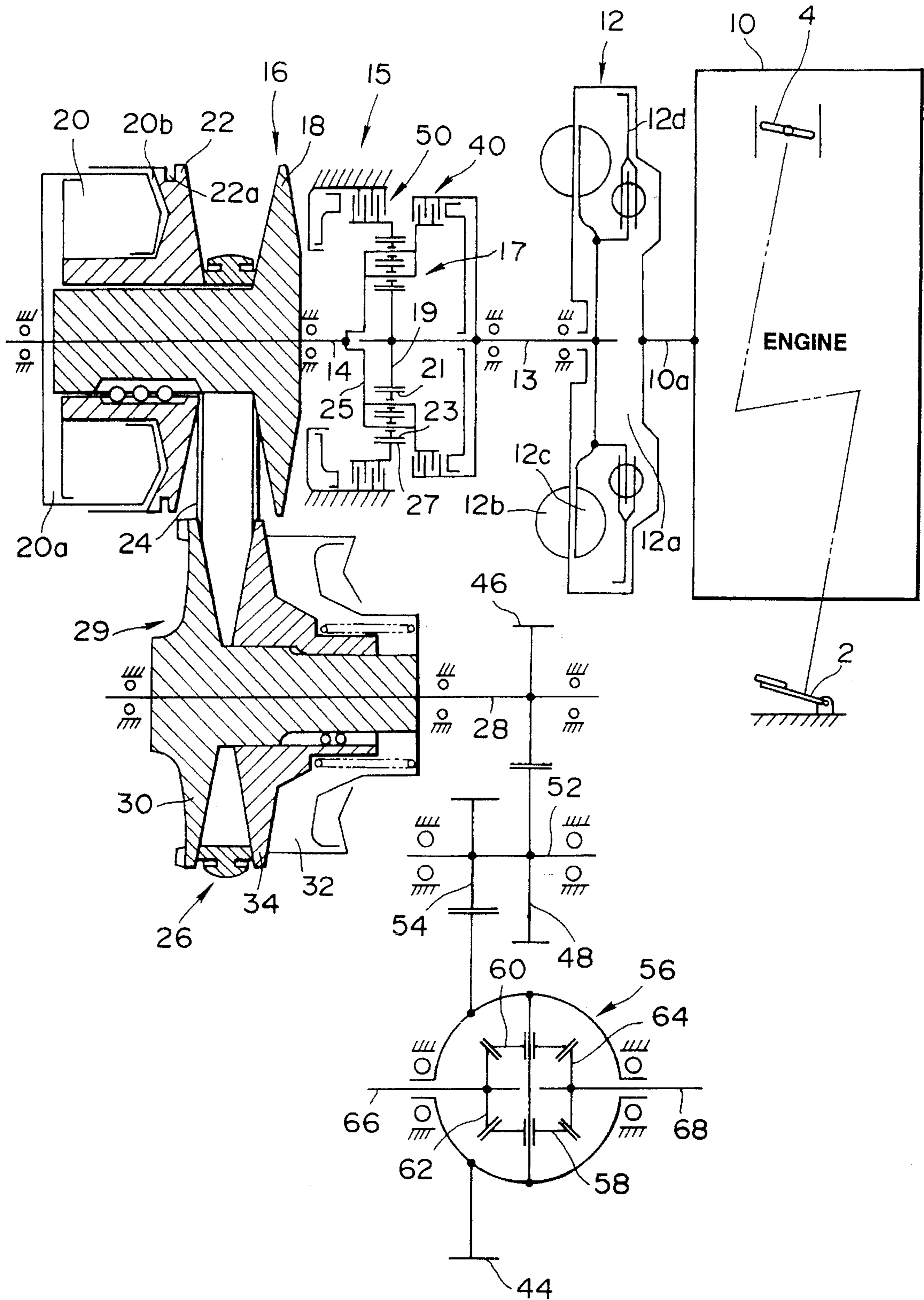


FIG. 1



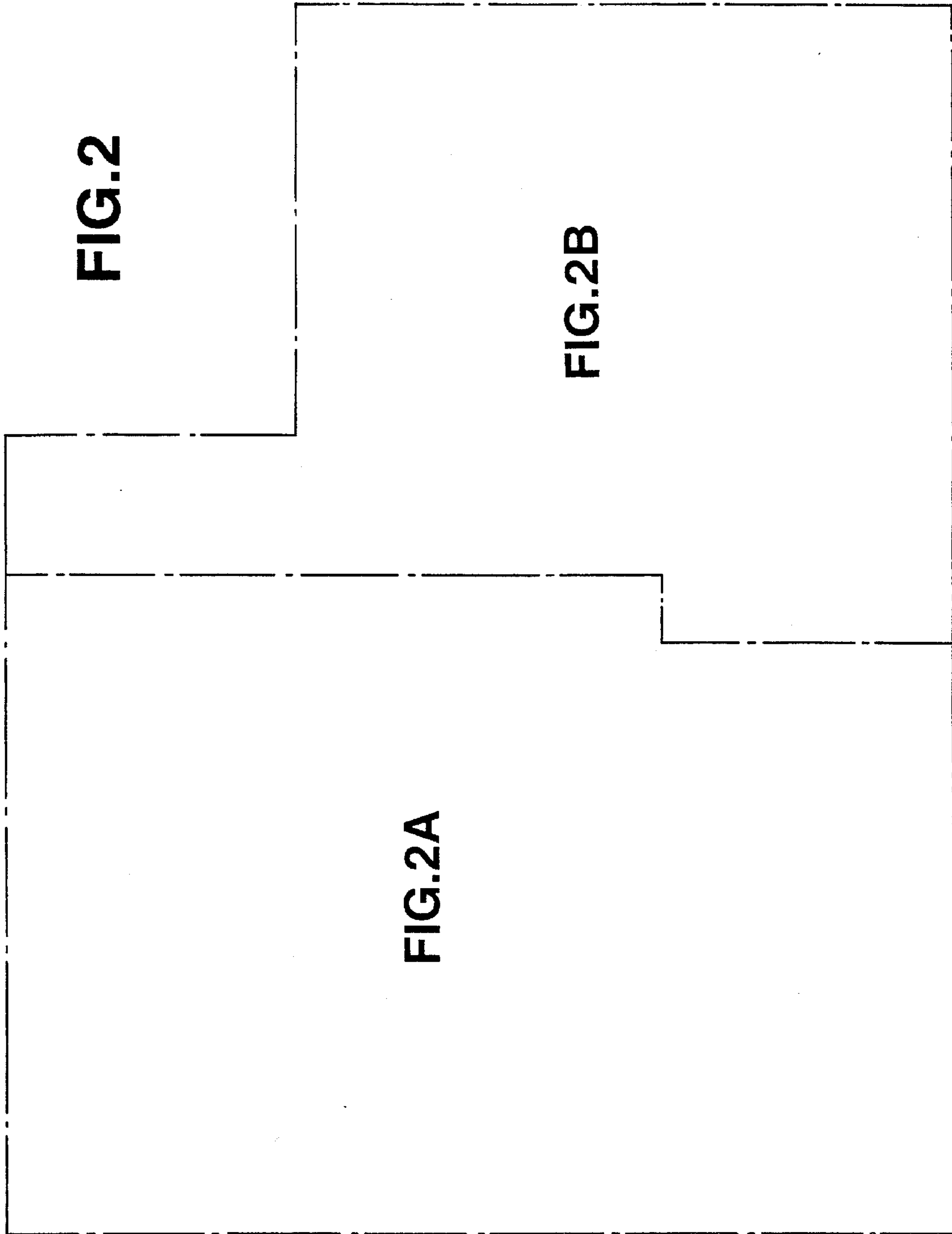


FIG.2A

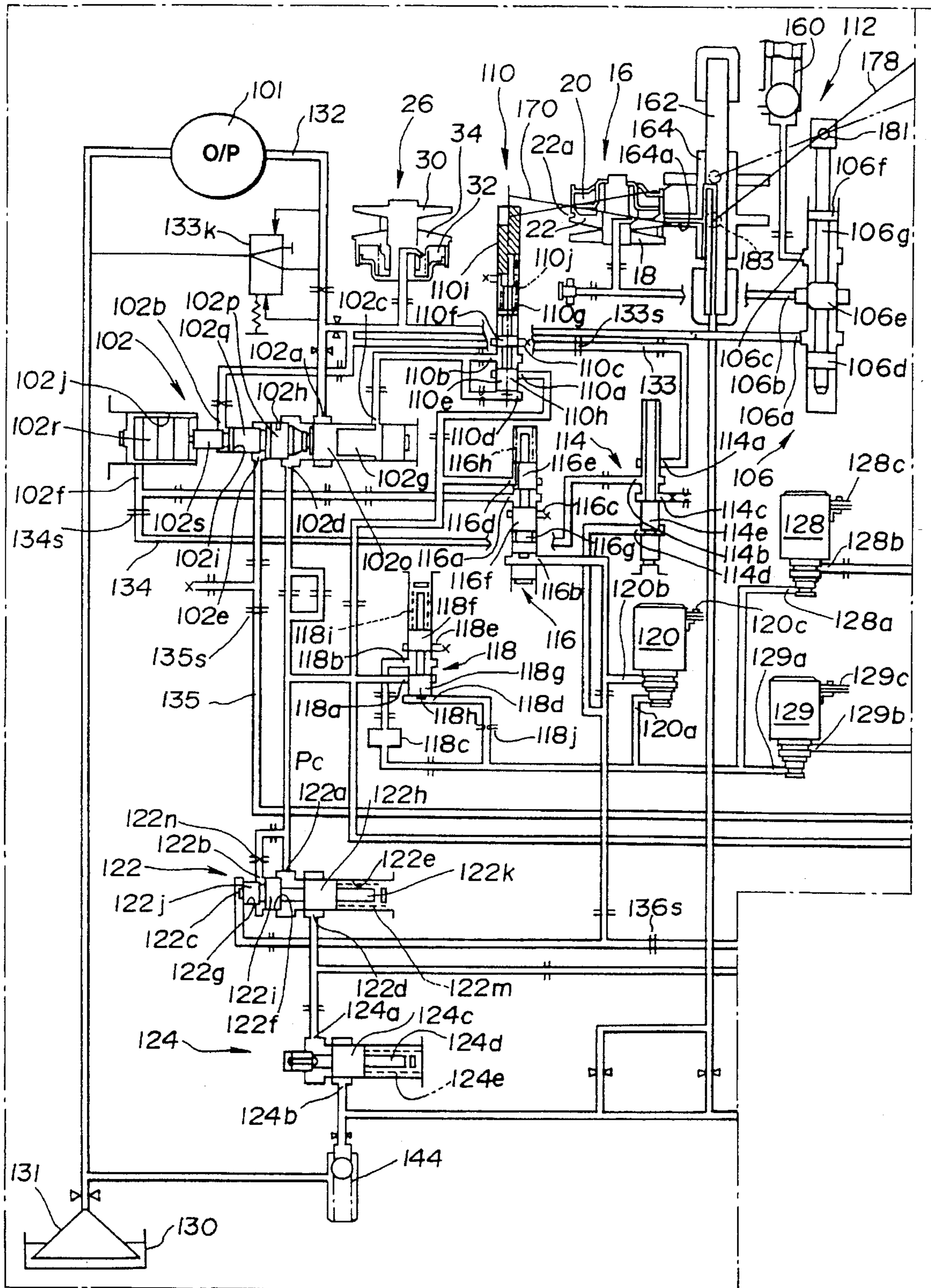


FIG.2B

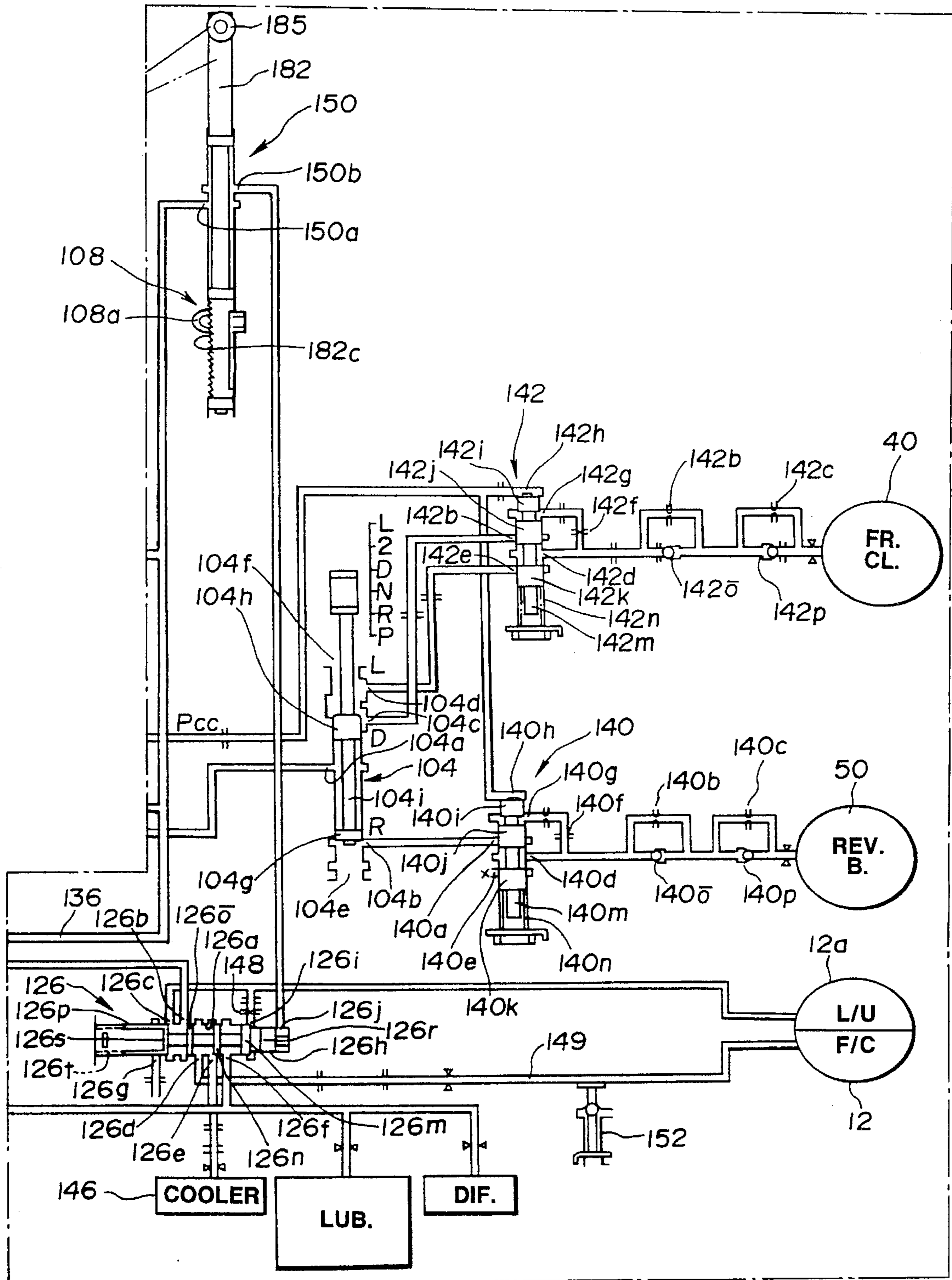


FIG. 3

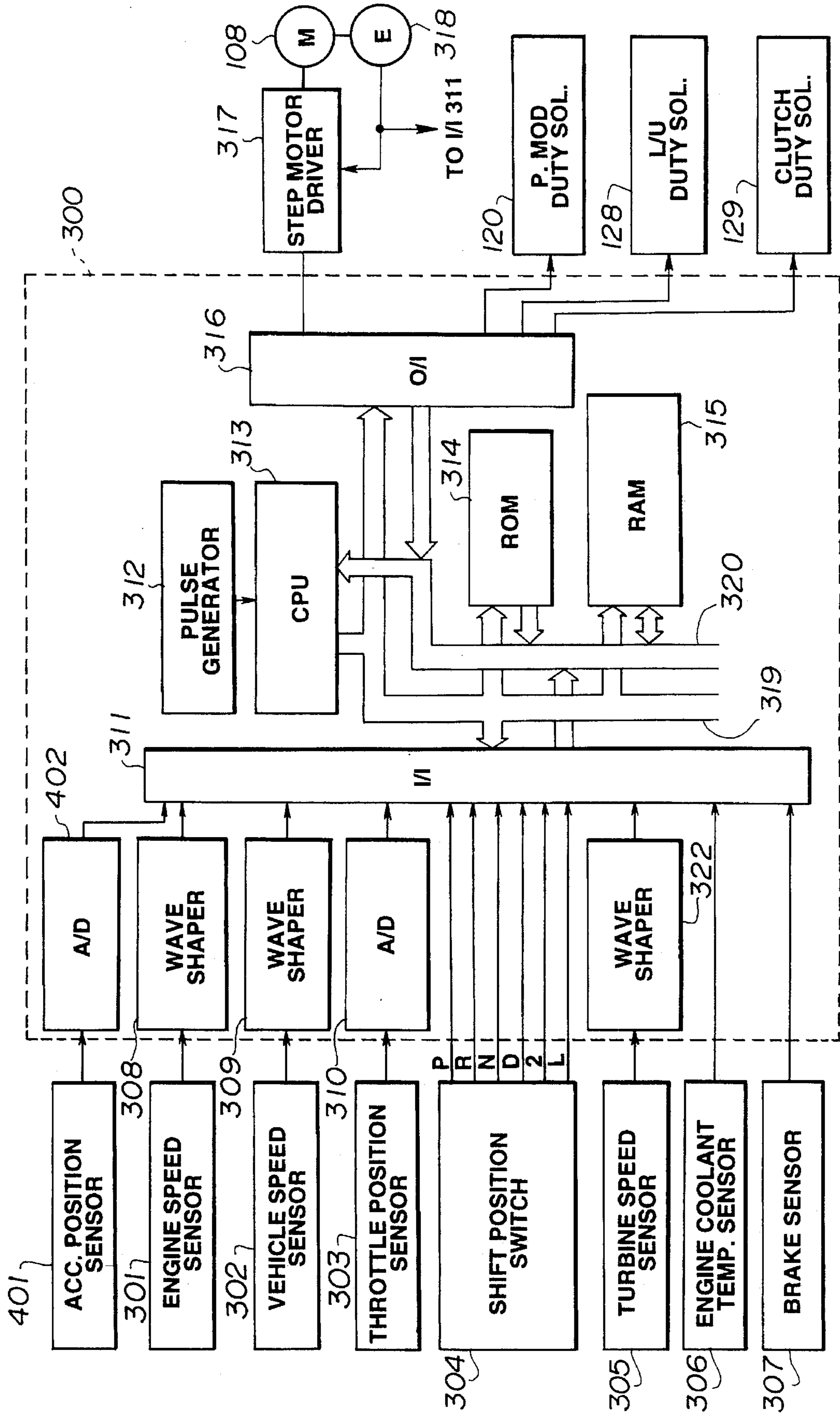


FIG.4

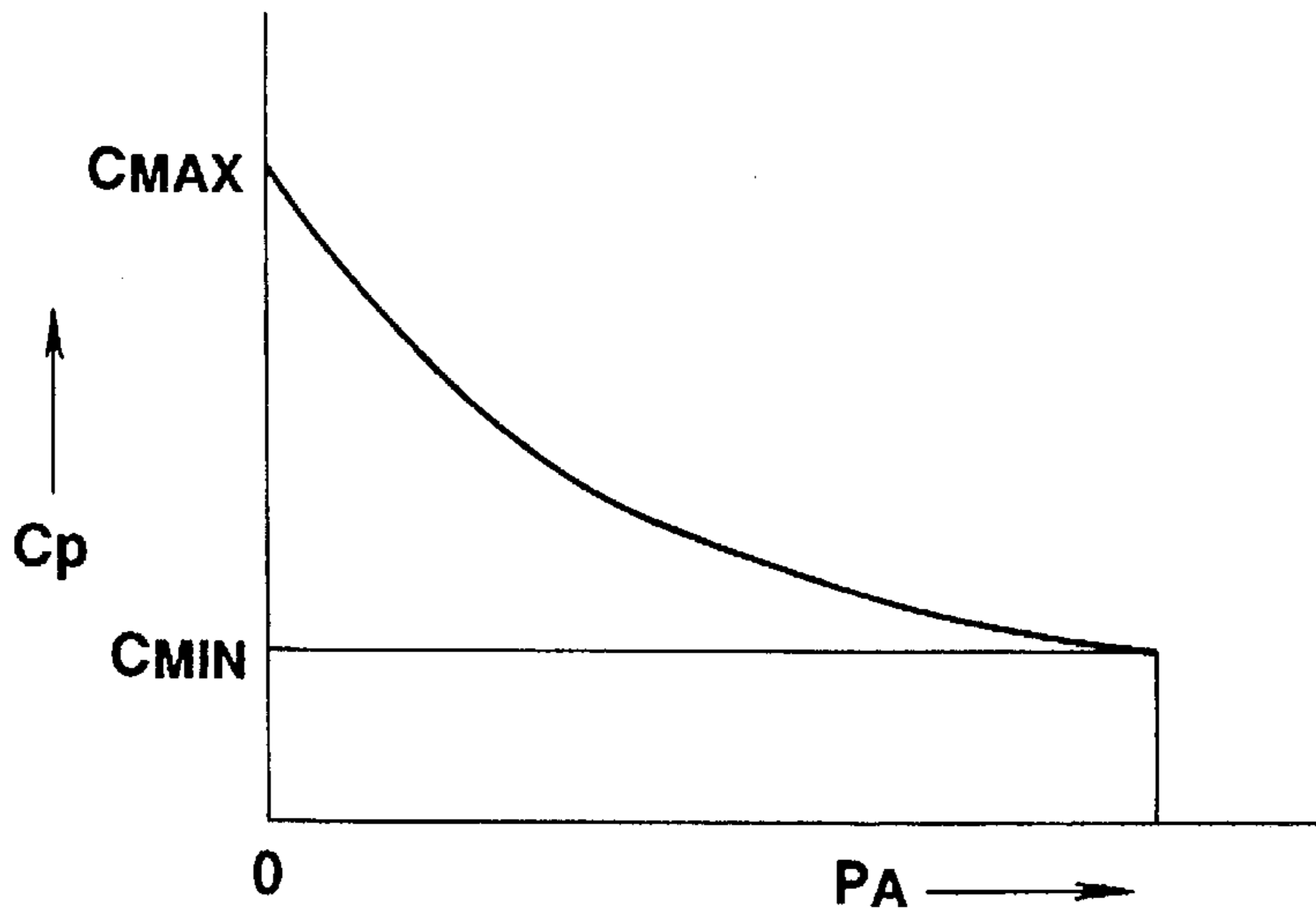


FIG.6

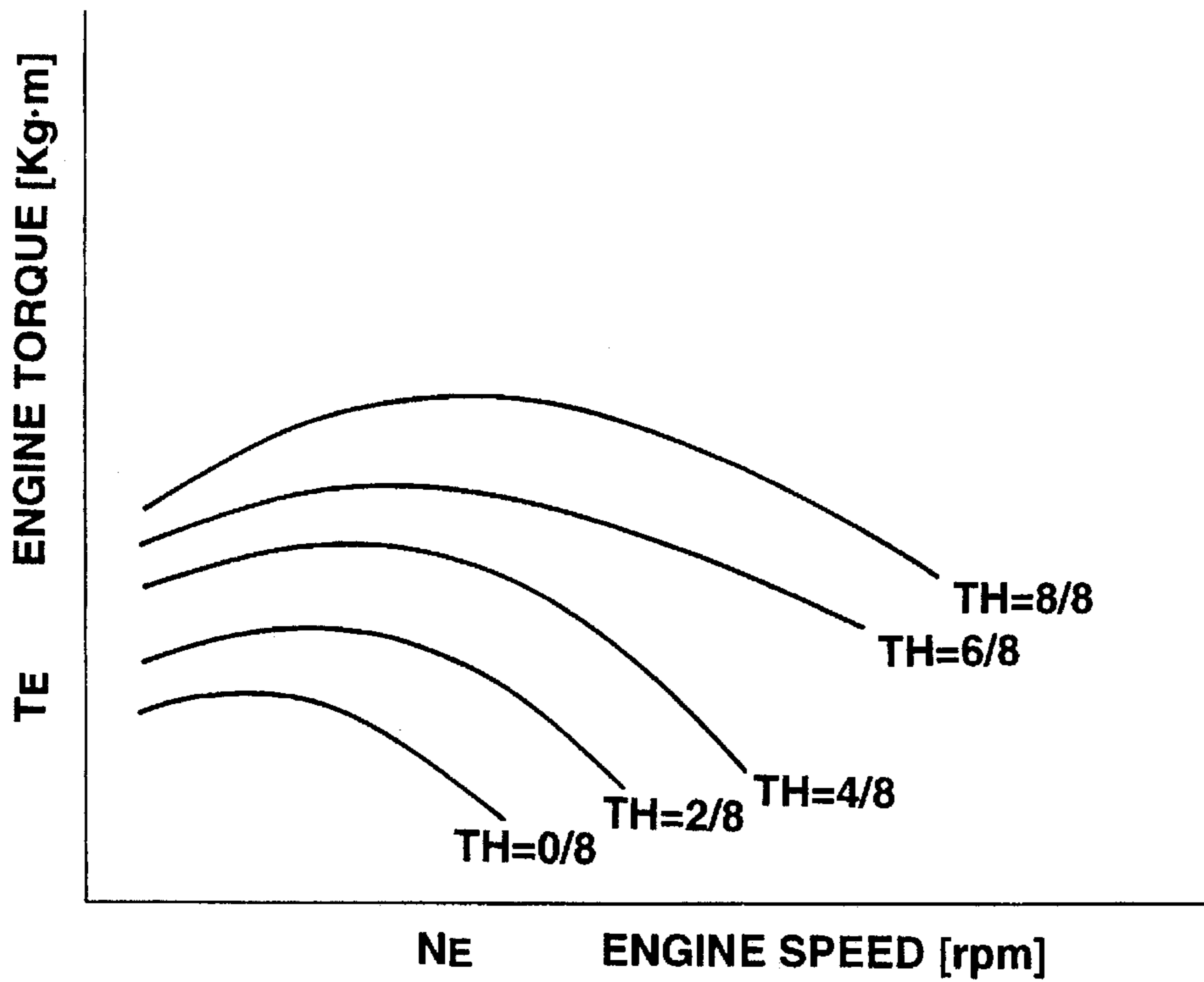


FIG. 5A

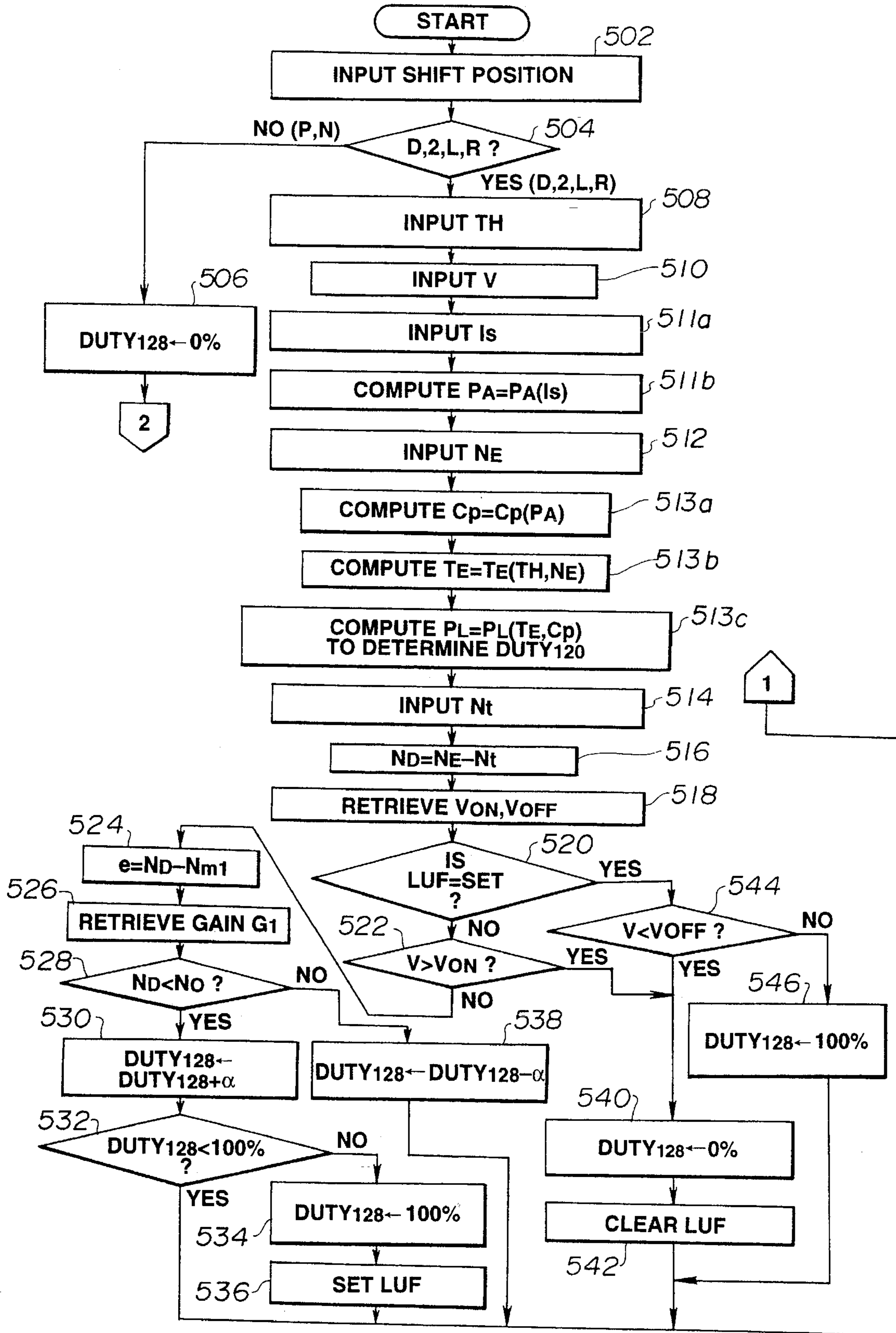


FIG. 5B

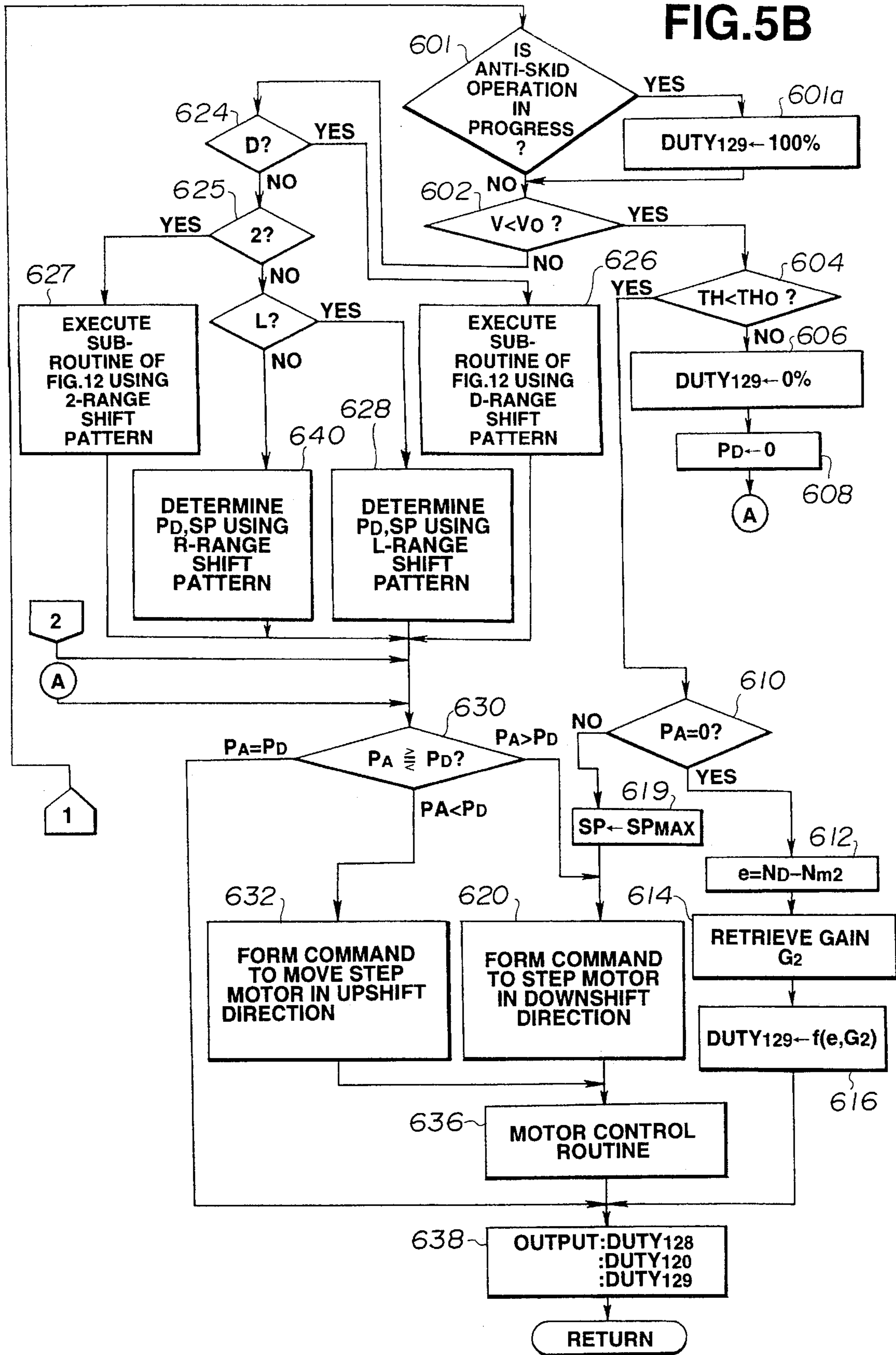


FIG.7

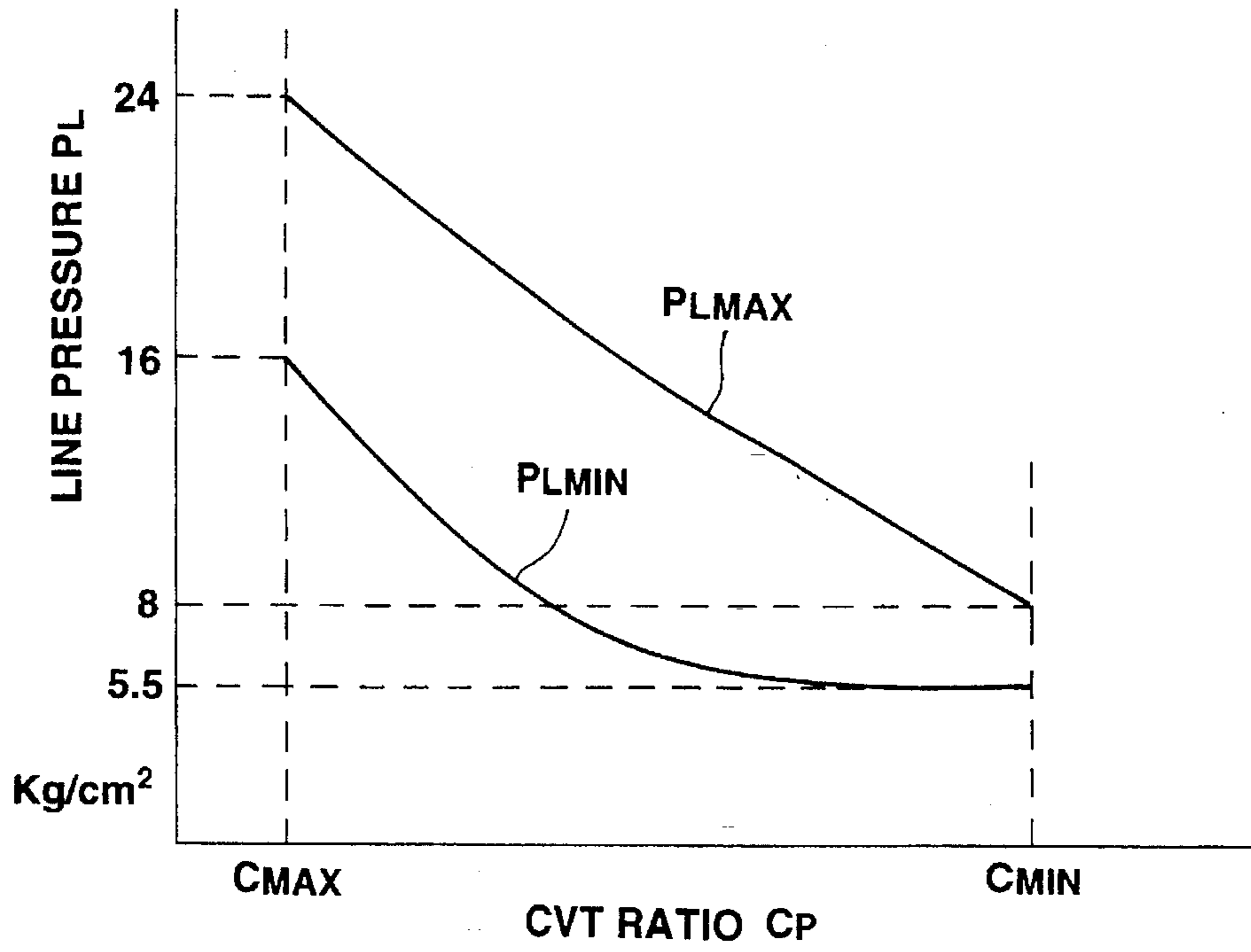


FIG.8

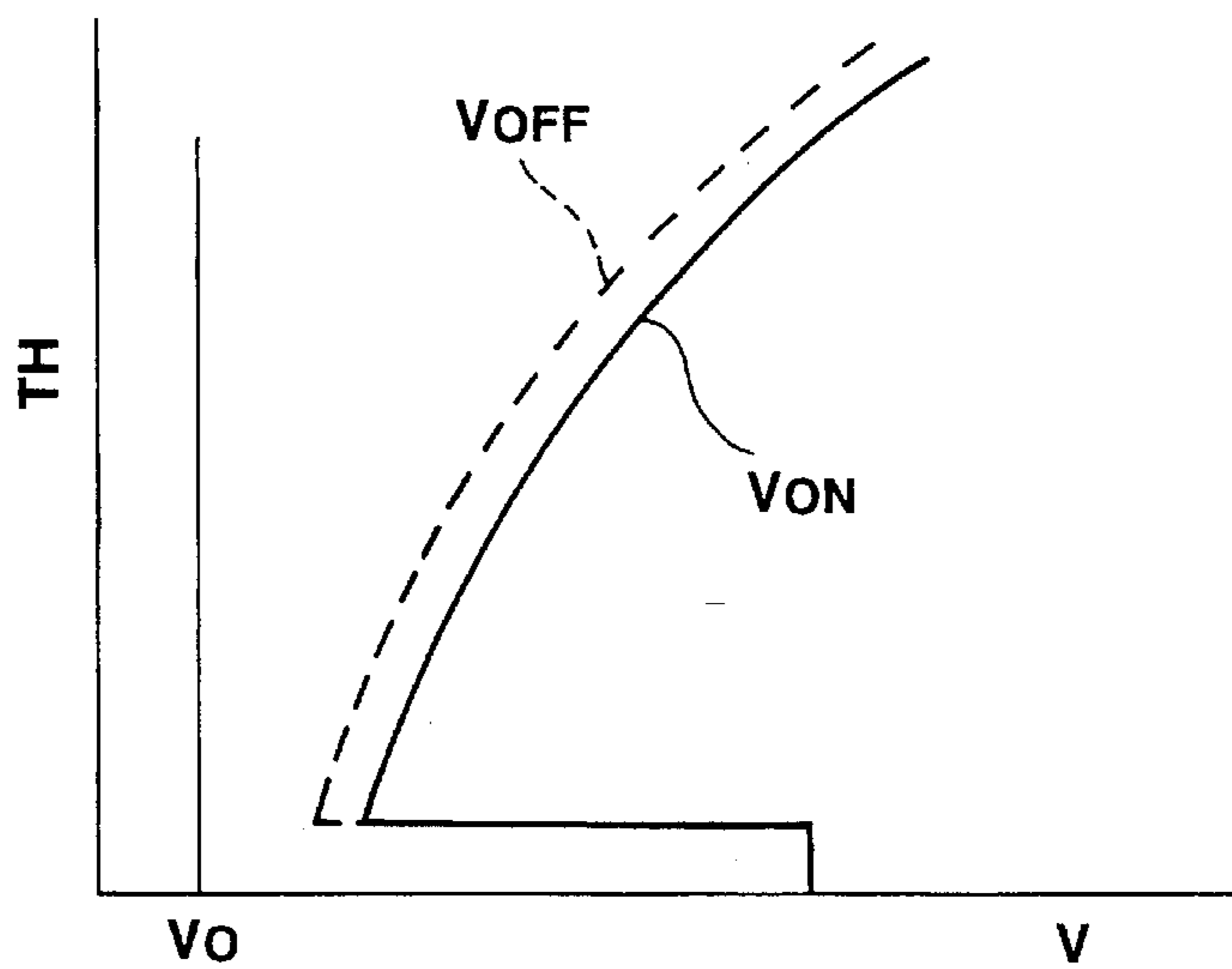


FIG.9

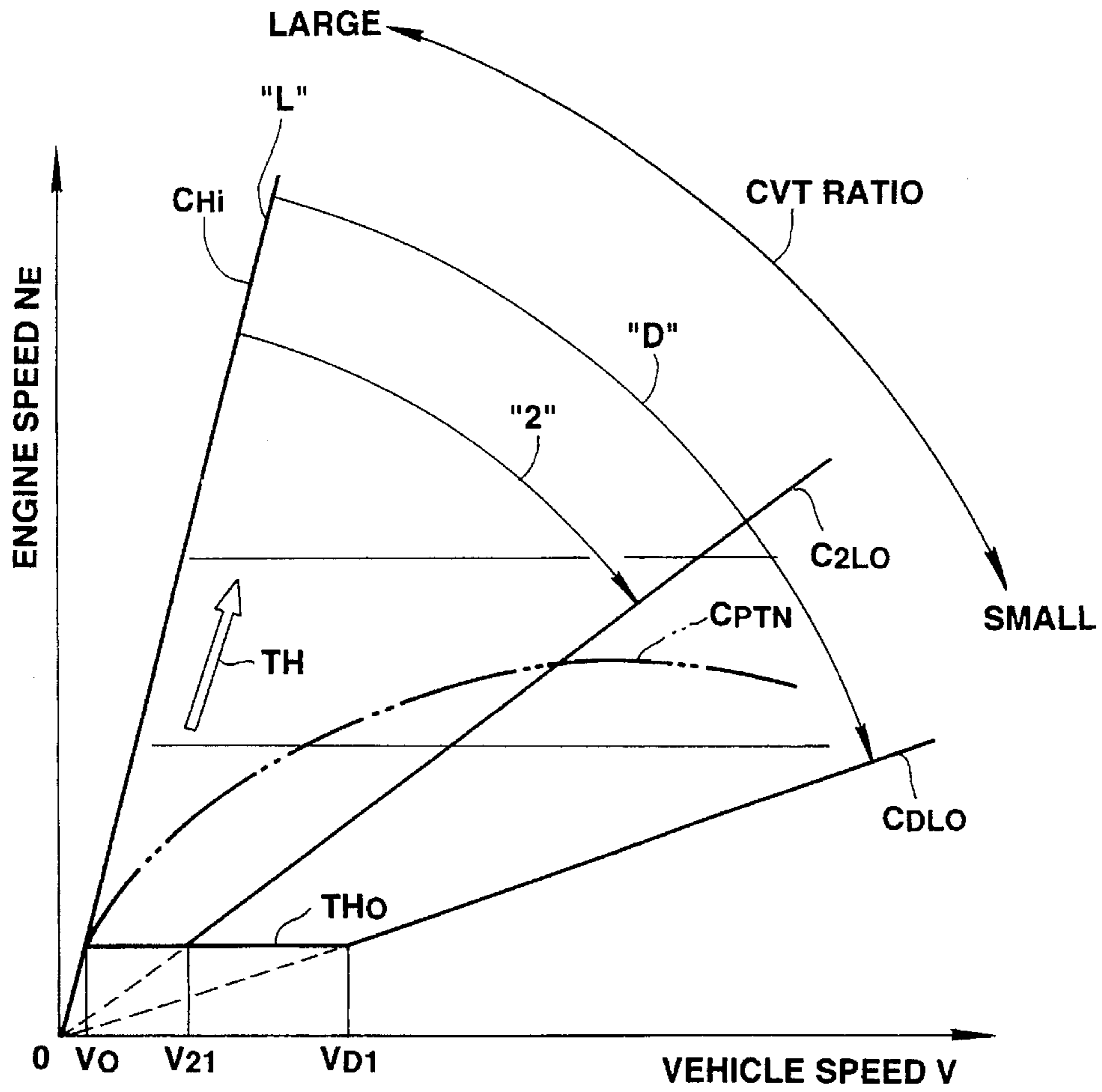


FIG.11

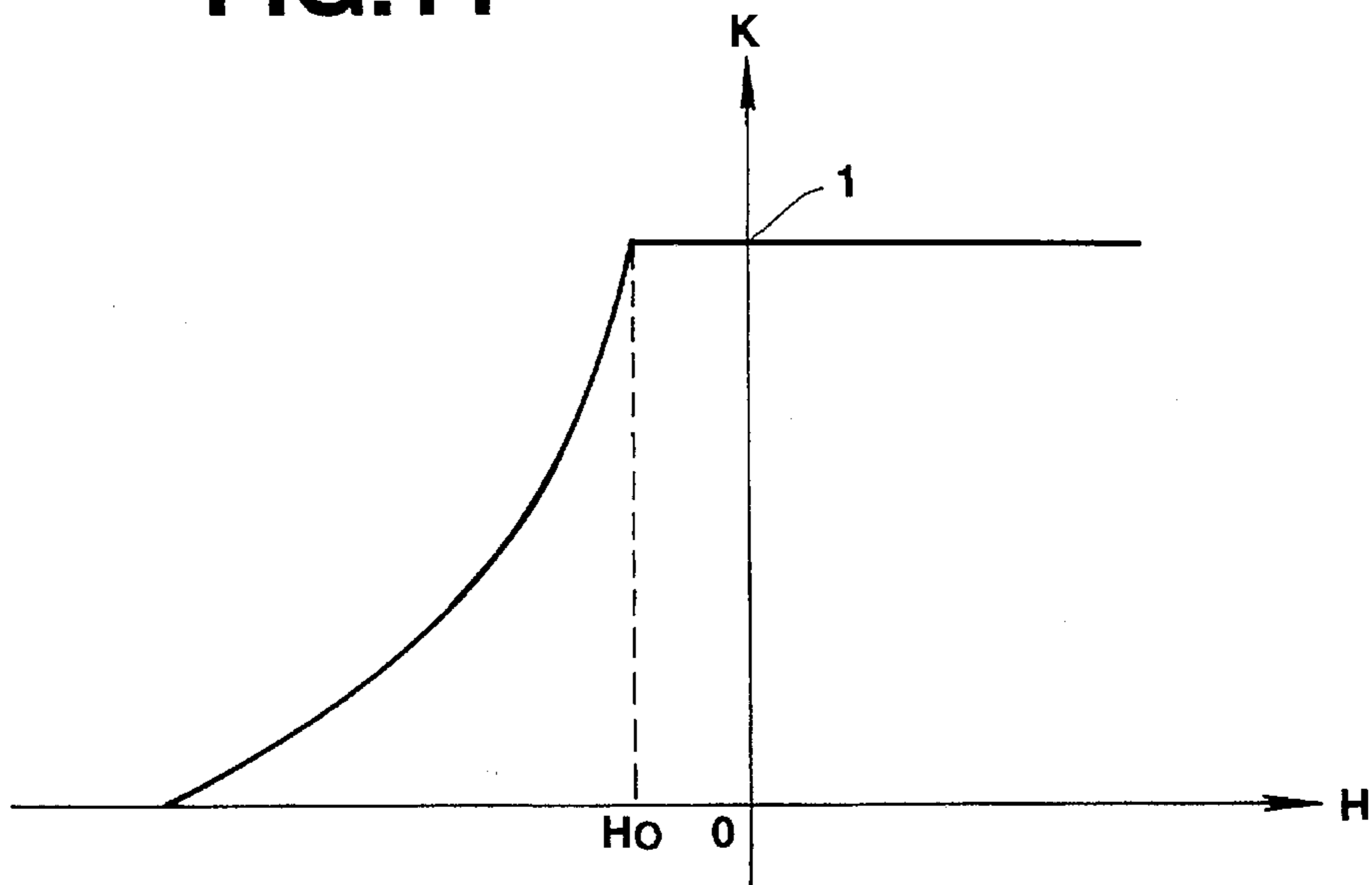


FIG.10

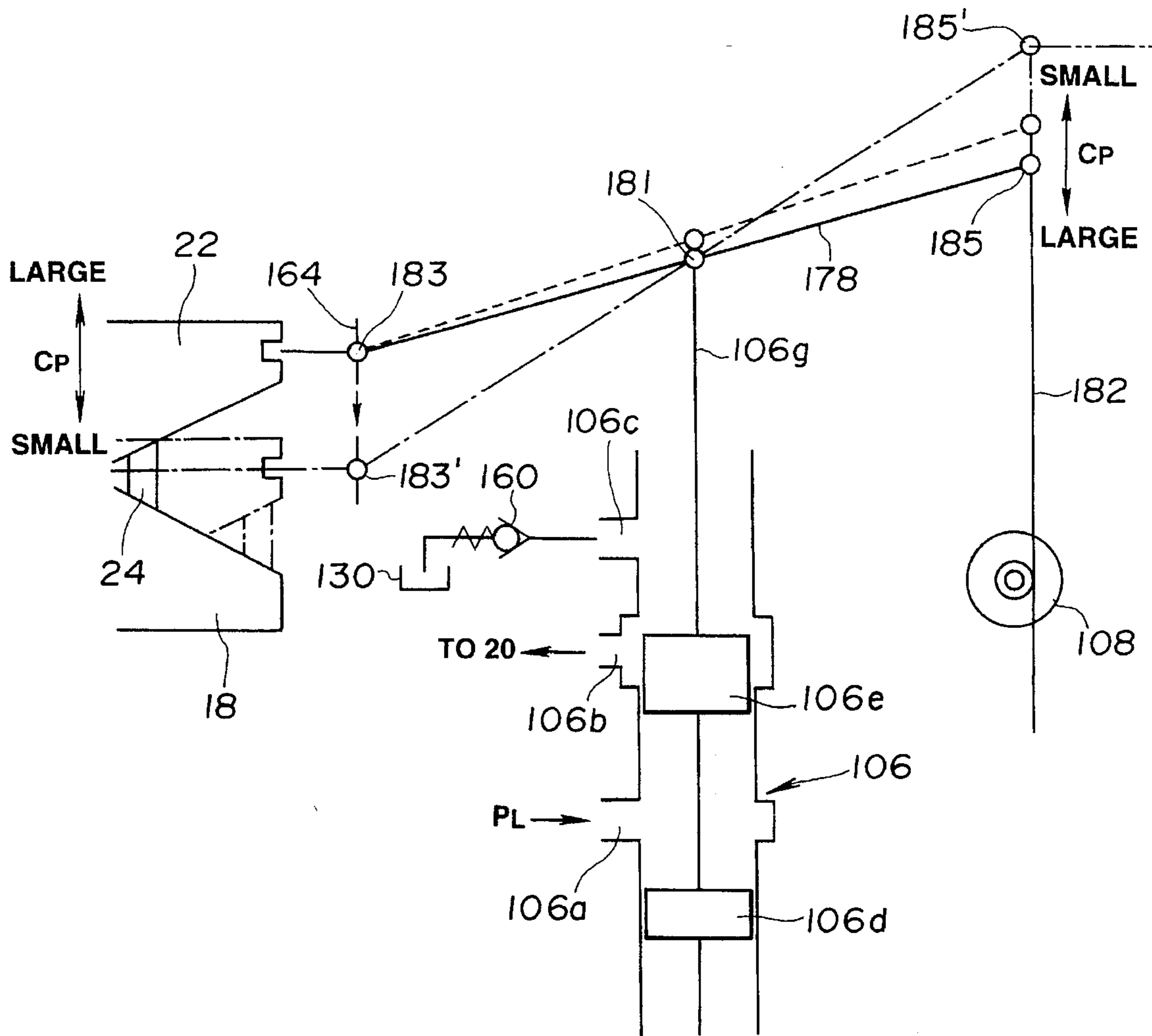


FIG.12

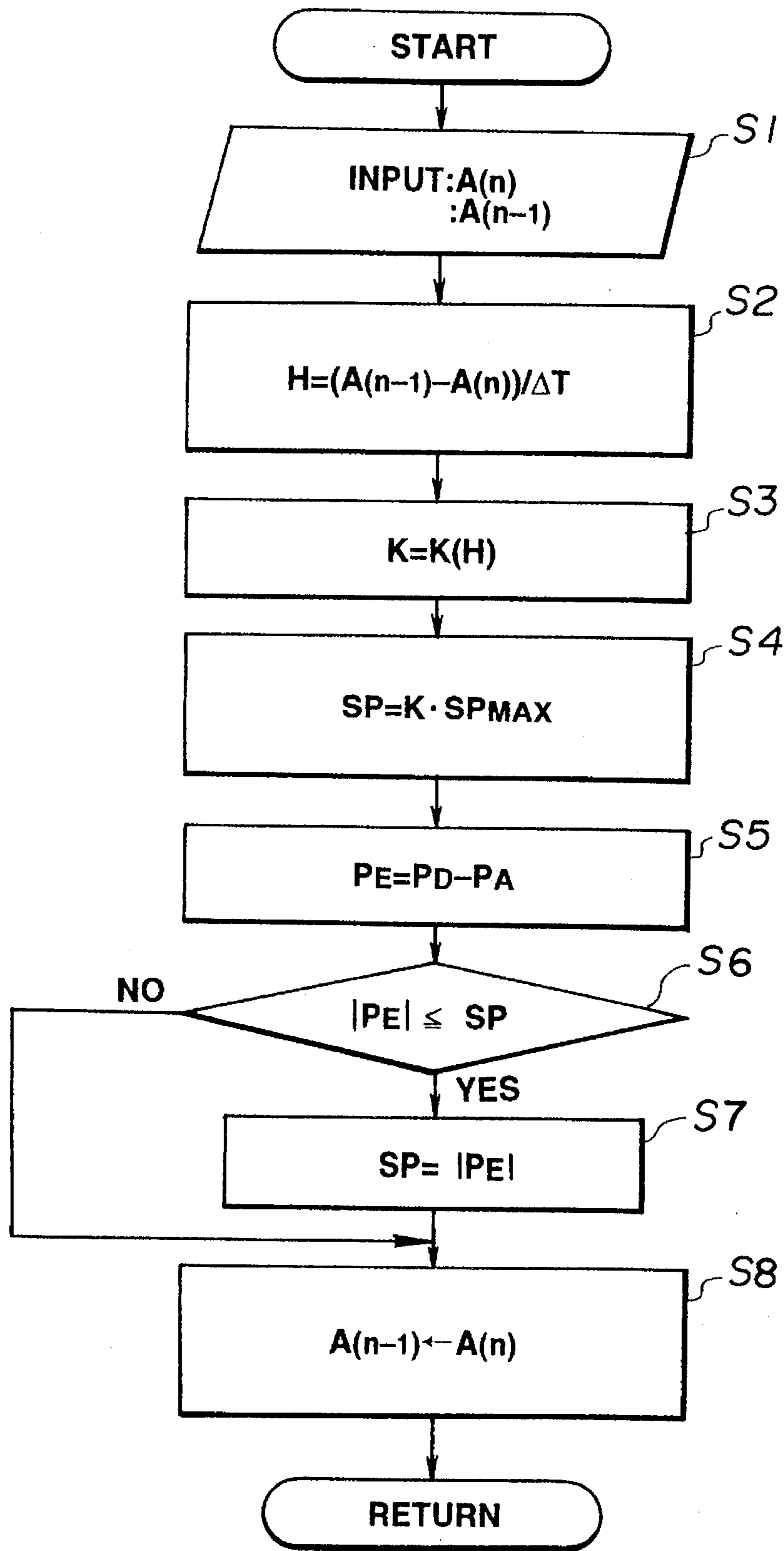
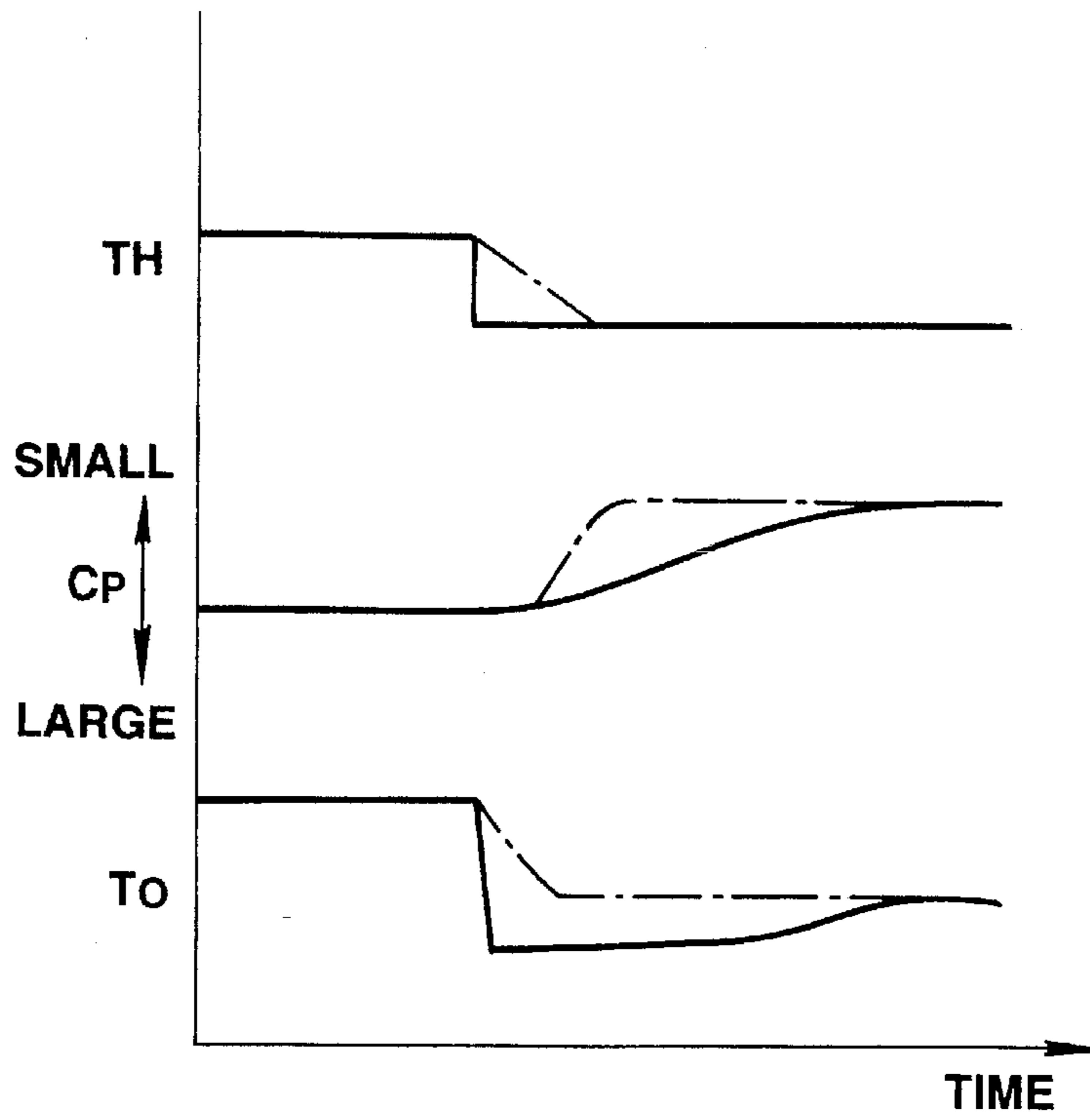
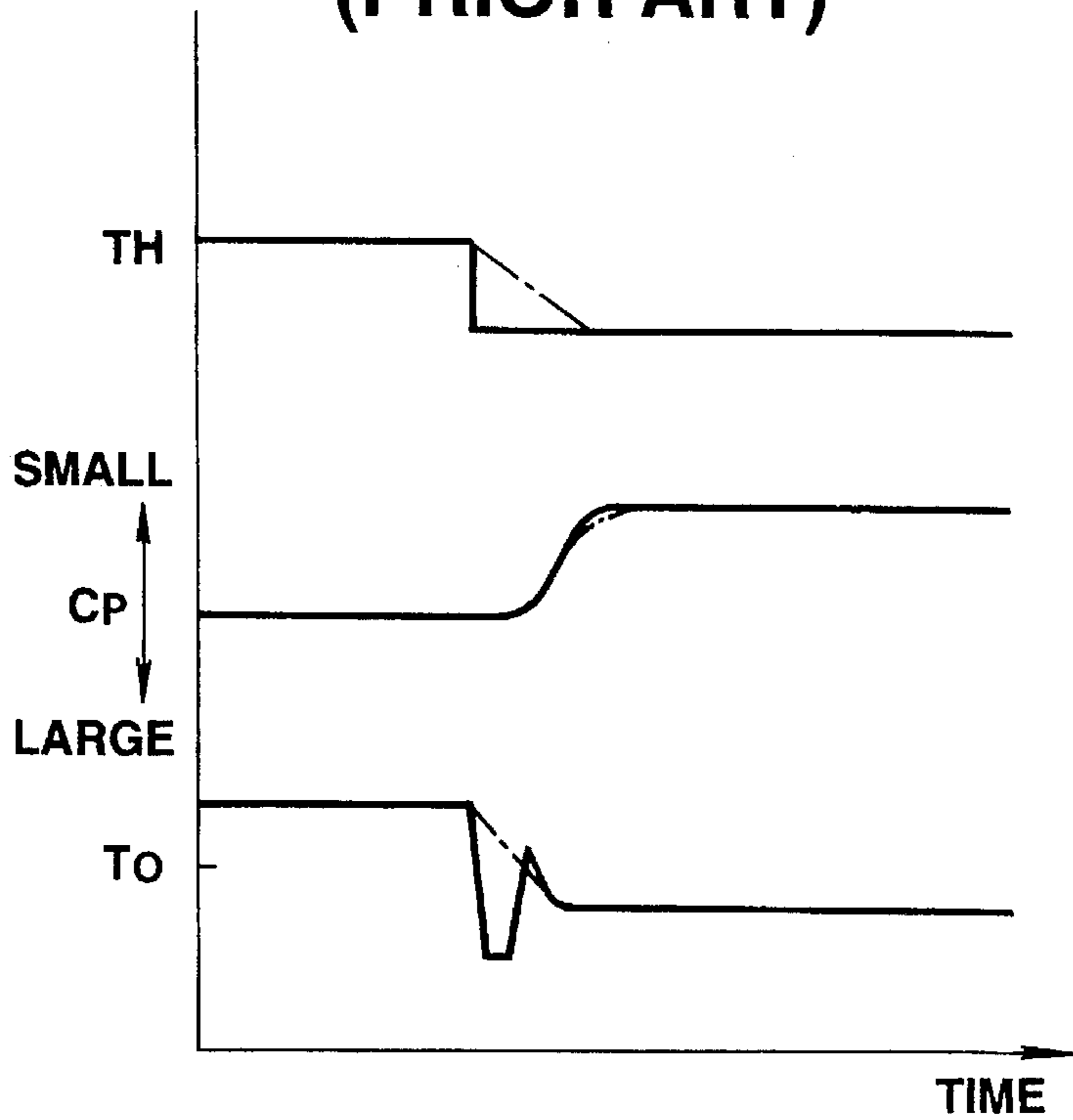


FIG. 13



**FIG. 14
(PRIOR ART)**



CVT RATIO RATE CONTROL AS A FUNCTION OF THROTTLE RELEASE RATE

BACKGROUND OF THE INVENTION

The present invention relates to a control of an automotive continuously variable transmission (CVT), and more particularly to a CVT ratio rate control.

Previously, there have been various methods to control the automotive CVT ratio. The most straight forward way is to define a desired CVT ratio in response to a vehicle speed and a power request command, such as a throttle opening degree or a depression of gas pedal, and then adjust an actuator, such as, a stepping motor, of a ratio control element of the CVT in a manner to achieve that CVT ratio. There occurs a change in desired CVT ratio toward the minimum or smallest ratio in response to a quick release of a gas pedal. Thus, the CVT ratio begins to change in upshift direction immediately after the operator has released the gas pedal quickly. This change in CVT ratio causes inertia of the engine, causing a peak in output torque of the CVT. Due to this torque peak, the operator experiences ride feel that the automotive vehicle is thrown forwardly or feels shocks. Besides, engine brake performance is not satisfactory.

What is desirable in driving an engine-CVT system is CVT ratio control dependent on the rate of change per unit time of the position of a gas pedal toward a released position thereof.

SUMMARY OF THE INVENTION

The invention uses CVT ratio rate control as a means of controlling a speed of change in CVT ratio in upshift direction toward the minimum or smallest ratio in response to the rate of change per unit time of the position of a gas pedal toward a released position thereof. This type of control thus enables control of CVT ratio for improved engine brake performance in response to the rate of change per unit time of the position of a gas pedal toward a released position thereof.

According to the present invention, there is provided a control system, for a vehicle drivetrain with an engine with a gas pedal and a continuously variable transmission (CVT) including a ratio control element positionable to various command positions in response to the ratio rate determined by controller means to establish various CVT ratios, wherein the controller means alters the rate of change per unit time of CVT ratio in response to the rate change per unit time of the position of a gas pedal toward a released portion thereof.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic view of a continuously variable transmission (CVT) incorporated in a transaxle drivingly connected to an engine;

FIG. 2 illustrates how FIG. 2A and FIG. 2B are combined;

FIGS. 2A and 2B, when combined, are a hydraulic circuit of a hydraulic control system;

FIG. 3 is a block diagram of a CVT controller;

FIG. 4 shows, in the fully drawn line, a C_P (CVT ratio) versus P_A (actual step number) characteristic curve;

FIGS. 5A and 5B, when combined, are a flow diagram of a main control routine responsible for line pressure control, lock-up control and creep control;

FIG. 6 shows a family of T_E (engine torque) versus N_E (engine speed) characteristic curves in response to TH (throttle opening degree) of the engine;

FIG. 7 shows a curve P_{LMAX} interconnecting plots of the maximum value of line pressure P_L at each CVT ratio C_P with the highest value of engine torque T_E , and another curve P_{LMIN} interconnecting plots of the minimum value of line pressure P_L at each CVT ratio with the lowest value of engine torque T_E ;

FIG. 8 shows a lock-up on vehicle speed V_{ON} and a lock-up off vehicle speed V_{OFF} along with a threshold vehicle speed V_0 of ratio control;

FIG. 9 is a graphical representation of extensions, in CVT ratio, of D and 2 ranges in comparison with fixed CVT ratio for L range;

FIG. 10 is a diagram illustrating relation between a stepping motor, a ratio control element, a ratio control valve and a driver pulley;

FIG. 11 is a graphical representation of characteristic of coefficient K;

FIG. 12 is a flow diagram illustrating a sub-routine;

FIG. 13 is a time chart illustrating variation in output shaft torque T_O , according to the present invention, upon or immediately after a quick drop in throttle opening degree TH; and

FIG. 14 is a time chart illustrating a peak produced in output shaft torque T_O , according to the afore-discussed prior art, upon or immediately after a quick drop in throttle opening degree TH.

DETAILED DESCRIPTION OF THE INVENTION

Referring to FIG. 1, a block 10 represents an engine. The engine 10 has a throttle 4 which opens in degrees in response to depressed position of a gas pedal 2. The engine 10 is drivingly connected to a hydrokinetic torque transmitting unit in the form of a fluid coupling 12 or a torque converter if desired. The fluid coupling 12 is drivingly connected to a continuously variable transmission (CVT) 29 through a forward/reverse change-over mechanism 15 including a planetary gearing 17, a forward clutch 40 and a reverse brake 50. This drive connection can be interrupted when a forward clutch 40 is disengaged with a reverse brake 50 released.

For forward drive, the forward clutch 40 is engaged with the reverse brake 50 released, while for reverse drive, the reverse brake 50 is applied with the forward clutch 40 disengaged. The CVT 29 is drivingly connected to a final drive 56. The CVT 29 includes a driver pulley 16, a follower pulley 26 and a V-belt 24. It will be understood that the present invention is not limited to the use of CVT's with pulley V-belt power transfer. The present invention is operational with any other type of CVT in which the ratio can be controlled, including traction drive type CVT's.

Detailed description of FIG. 1 is found in U.S. Pat. No. 5,067,372 issued on Nov. 26, 1991 to Suzuki and commonly assigned herewith. This U.S. Pat. corresponds to DE-A 40 25 455. For further understanding of FIG. 1, reference should be made to description of FIG. 1 of this U.S. Pat. No. 5,067,372 which is hereby incorporated by reference.

Referring to FIGS. 2A and 2B, a hydraulic control system for the transaxle including the CVT 29 is described.

An oil pump 101 has a suction side to draw in oil from a tank 130 via a strainer 131, and a discharge side connected

to one end of a passage 132. The other end of the passage 132 is connected to a port 102a of a line pressure regulator valve 102. This passage 132 has a branch connected to a cylinder chamber 32 of the follower pulley 26 of the CVT 29, a branch connected to a port 102b of the line pressure regulator valve 102, and a branch connected to a port 106a of a shift control valve 106. The passage 132 has a branch 133 connected to a port 114a of a transfer valve 114. In this embodiment, the branch 133 is blocked by a separator 133s.

In the line pressure regulator valve 102, pressure regulation is effected to generate a line pressure. The line pressure builds up not only at the port 102a, but also at a feedback port 102b. During the pressure regulation, oil under pressure is delivered from a port 102d to ports 122a and 122b of a clutch relief valve 122, a port of a pilot valve 118, a port 104a of a manual selector valve 104, a port 110a of a CVT ratio pressure transducer valve 110, and a port 116d of a pressure modifier valve 116. A line pressure relief valve 133k is schematically shown and connected to the passage 132 to suppress an excessive increase of line pressure.

In addition to the inlet port 118a connected to the port 102d of the line pressure regulator valve 102, the pilot valve 118 has an outlet port 118b, a feedback port 118d and a drain port 118e. Fluidly disposed between the outlet port 118b and feedback port 118d is a filter 118c. Fluidly disposed between the filter 118c and the feedback port 118d is an orifice 118j. The pilot valve 118 has a spool 118h with two lands 118f and 118g and a spring 118i biasing the spool 118h toward the feedback port 118d. The spool 118h is subjected to pressure at the feedback port 118d. If this pressure is zero, the port 118a is uncovered by the land 118g, while the drain port 118e is covered by the land 118f, causing an increase in pressure at the outlet port 118b. This increase in pressure is transmitted to the feedback port 118d, moving the spool 118h against the spring 118i. During this movement of the spool 118h, the land 118g covers the inlet port 118a, while the land 118f uncovers the drain port 118e, causing a drop in pressure at the outlet port 118b. This drop in pressure allows the spring 118i to move the spool 118h toward the feedback port 118d. During this movement of the spool 118h, the land 118f covers the drain port 118e, while the land 118g uncovers the inlet port 118a, causing an increase in pressure at the outlet port 118b. As a result, the spool 118h assumes an equilibrium position to produce a constant pressure determined by the bias force of the spring 118i.

This constant pressure, which may be called a pilot pressure, is supplied to an inlet port 120a of a pressure modifier duty solenoid valve 120, an inlet port 128a of a lock-up duty solenoid valve 128, and an inlet port 129a of a clutch duty solenoid valve 129. The pressure modifier duty solenoid valve 120 is responsible for line pressure control. The lock-up duty solenoid valve 128 is responsible for lock-up control. The clutch duty solenoid valve 129 is responsible for creep control.

In addition to the inlet port 120a, the pressure modifier duty solenoid valve 120 has a drain port 120c and an outlet port 120b connected to port 116b of the pressure modifier valve 116. When duty ($DUTY_{120}$) of the solenoid valve 120 is 0%, a needle closes the drain port 120c, allowing the full supply of the pilot pressure from the inlet port 120a to the outlet port 120b. When the duty $DUTY_{120}$ is 100%, the needle opens the drainport 120c, draining oil from the outlet port 120b. The duty $DUTY_{120}$ can be adjusted to any value falling in a range from 0% to 100%. Pressure appearing at the outlet port 120b, which may be called a modifier valve control pressure, varies continuously between the highest level as high as the pilot pressure and the lowest level as

$DUTY_{120}$ varies continuously between 0% and 100%. In other words, the modifier valve control pressure is inversely proportional to $DUTY_{120}$.

In addition to the port 116 receiving the modifier valve control pressure from the solenoid valve 120, the pressure modifier valve 116 has a drain port 116c and an outlet port 116a connected to a port 102f of the line pressure regulator valve 102. The pressure modifier valve 116 has a spool 116g with two lands 116e and 116f and a spring 116h biasing the spool 116g. The land 116e is formed with an axial through orifice which provides constant fluid communication between a chamber accommodating therein the spring 116h and an annular chamber defined between the lands 116e and 116f. The outlet port 116a is always open to this annular chamber so that the pressure at the port 116a is transmitted through the axial orifice to the spring accommodating chamber to act on the land 116e in a direction assisting the action of the spring 116h. When the modifier valve control pressure supplied from the solenoid valve 120 to the port 116b is zero, the spool 116g assumes a spring set position where the land 116e covers the port 116d and the land 116f uncovers the drain port 116c. Under this condition, the outlet port 116a is drained. An increase in modifier valve control pressure moves the spool 116g against the spring 116h. During this movement of the spool 116g, the land 116e uncovers the inlet port 116d after the land 116f has covered the drain port 116c. Immediately after the inlet port 116d is uncovered to communicate with the outlet port 116a, there is a pressure increase at the outlet port 116a and also in the spring accommodating chamber. This pressure increase at the outlet port 116a moves the spool 116g in the direction assisting the action of the spring 116h, causing the land 116e to cover the inlet port 116d and the land 116f to uncover the drain port 116c, resulting in a decrease in pressure at the outlet port 116a. As a result, the spool 116g assumes an equilibrium position to produce, at the outlet port 116a, a pressure, which may be called a modifier pressure, variable with the modifier valve control pressure that is variable with the duty $DUTY_{120}$ of the solenoid valve 120.

The modifier pressure is supplied to the port 102f of the line pressure regulator valve 102. In addition to this port 102f and the before mentioned ports 102a, 102b and 102d, the line pressure regulator valve 102 has a port 102e which is drained and a port 102c. The line pressure regulator valve 102 has a spool 102s with lands 102o, 102p, 102q and 102r. The land 102r has the largest diameter and is slidably disposed in a bore section 102j having the largest diameter. Among the other three lands, the land 102o has the largest diameter, land 102h has a diameter smaller than that of the land 102o, and land 102s has the smallest diameter. These three lands 102o, 102p and 102q are slidably disposed in the bore sections 102g, 102h and 102i, respectively. The ports 102a and 102c are open to the bore section 102g, the ports 102d and 102e are open to the bore section 102h and port 102f is open to the bore section 102j. The spool 102s is biased by spring means, not illustrated, in a leftward direction, as viewed from FIG. 2A. In this embodiment, the port 102e is used as a drain port. If desired, this port 102e may be connected via a branch 135 to the lock-up duty solenoid valve 128. The branch 135 is blocked, in this embodiment, by a separator 135s. The setting of the line pressure regulator valve 102 is such that, with the same pressure supplied to the port 102c, as the modifier pressure at the port 102f decreases, the line pressure P_L increases from a minimum value P_{LMIN} (see FIG. 7) toward a maximum value P_{LMAX} . Thus, the line pressure P_L can be adjusted to any value between the minimum and maximum values P_{LMIN} and P_{LMAX} inclusive

by varying the duty $DUTY_{120}$ of the solenoid valve 120. In this embodiment, the pressure supplied to the port 102c varies with varying CVT ratio C_p . As shown in FIG. 7, as the CVT ratio C_p varies in a downshift direction from the minimum or smallest ratio C_{MIN} toward the maximum or largest ratio C_{MAX} , the maximum value P_{LMAX} of line pressure P_L increases at substantially the same rate, while the minimum value P_{LMIN} increases at a progressively increasing rate.

The modifier valve control pressure produced by the pressure modifier duty solenoid valve 120 is also supplied to a port 114d of the transfer valve 114 and a port 122c of the clutch relief valve 122. As mentioned before, the port 114a of the transfer valve 114 is connected to the branch 133 which is blocked by the separator 133s. the transfer valve 114 has a port 114b and a drain port in addition to the ports 114a and 114d. The port 114b is connected through a passage 134 to the port 102f of the line pressure regulator valve 102. But, this passage 134 is blocked by a separator 134s. Thus, the transfer valve 114 does not play any role in this control system owing to the separators 133s and 134s. The transfer valve 114 has a spool 114e which is moved by the modifier valve control pressure supplied to the port 114d to cover the drain 114c, enabling supply of pressure from the port 114a to the port 114b.

The pressure supplied to the port 102c is generated by the CVT ratio pressure transducer valve 110. This pressure may be called as a CVT ratio dependent pressure since a sensor shoe 164 movable with a positionable pulley half 22 of the driver pulley 16 is connected through a lever 170 to a movable spring retainer 110i to change bias force of a spring 110j in response to a position of the pulley half 22. The sensor shoe 164 is non-rotatably but slidably supports a guide 162 and has a radially extending flange 164a engaged in groove 22a of the pulley half (see FIGS. 1 and 10) to follow movement of the pulley half 22. The lever 170 is connected to the sensor shoe 164 at one end and to the spring retainer 110i at the opposite end. At a point between the two ends, the lever 170 is pivoted. In addition to the port 110a, the CVT ratio pressure transducer valve 110 has a port 110b connected to the port 102c of the line pressure regulator valve 102, a feedback port 110d connected to the port 110b, and a drain port 110c. The transducer valve 110 has a spool 110h having three lands 110e, 110f and 110g. The spring 110j is disposed between the spring retainer 110i and the spool 110h. As the positional pulley half 22 moves away from the pulley half 18 to increase CVT ratio, the sensor shoe 164 moves the one end of the lever 170 and the opposite end of the lever 170 moves the spring retainer 110i toward the spool 110h to compress the spring 110j. The bias force of the spring 110j increases, causing the CVT ratio dependent pressure at the port 110b to increase accordingly.

In addition to the inlet port 106a receiving the line pressure, the shift control valve 106 has an outlet port 106b connected to a cylinder chamber 22 of the driver pulley 16, and a low pressure port 106c connected via a pressure maintaining or relief valve 160 to the tank 130. The shift control valve 106 has a spool 106g having three lands 106d, 106e and 106f. As viewed from FIG. 2A, upward movement of the spool 106g from the illustrated position causes the land 106 to increase fluid communication between the outlet port 106b and the inlet port 106a and decreases fluid communication between the outlet port and the low pressure port 106c, resulting in an increase in pressure within the cylinder chamber 22 of the driver pulley 16, moving the pulley halves 22 and 18 toward each other. Downward movement of the spool 106g from the illustrated position

causes the land 106 to decrease fluid communication between the outlet port 106b and the inlet port 106a and increases fluid communication between the outlet port and the low pressure port 106c, resulting in a decrease in pressure within the cylinder chamber 22 of the driver pulley 16, moving the pulley halves 22 and 18 apart.

A ratio control element is in the form of a rod or spool 182 of a shift command valve 150. The rod 182 is drivingly engaged via a rack 182c and a pinion 108a with an actuator in the form of a stepping motor 108.

In response to a ratio change requirement, the stepping motor 108 turns to move a lever 178, connected to the rod 182 turns to move a lever 178, connected to the rod 182 at a pin 185, to control the ratio rate of the CVT 29. The lever 178 is connected to the rod 182 at one end and to the sensor shoe 164 via a pin 183 at the other end. At a point between the two ends, the lever 178 is pivotably connected via a pin 181 to the spool 106g of the shift control valve 106. This arrangement, which may be called a shift operation mechanism 112, provides position feedback of the pulley 16 to the shift control valve 106. As the stepping motor 108 moves the lever 178, the shift control valve 106, in response to the movement of the valve spool 106g, alters the pressure supplied to the driver pulley 16 although the pressure supplied to the follower pulley 26 is unaltered. As the pressure supplied to the driver pulley 16 is altered, the pulley halves 18 and 22 move, changing the ratio of the CVT 29 at a rate dependent upon the speed of movement of the pulley halves 18 and 22. As the sensor shoe 164 moves with the pulley half 22, the lever 178 moves repositioning the valve spool 106g, providing a means for the shift control valve 106 to stop the positional pulley half 22 from moving.

The ratio rate of the CVT 29 is controlled by controlling the speed of the movement of the lever 178. The faster the stepping motor 108 turns to move the lever 178, the faster the CVT ratio changes.

In addition to the inlet port 122a, feedback port 122b and port 122c receiving the modifier valve control pressure, the clutch relief valve 122 has an outlet port 122d. The feedback port 122b is connected via an orifice 122n to the inlet port 122a. The outlet port 122d is connected to an inlet port 124a of a torque converter relief valve 124. The clutch relief valve 122 has three bore sections, namely the largest diameter bore section 122e formed with the inlet and outlet ports 122a and 122d, the smallest diameter bore section 122g formed with the port 122c and an intermediate diameter bore section 122f formed with the feedback port 122b. The intermediate bore section 122f is disposed between and interconnects the largest and smallest diameter bore sections 122e and 122g. The clutch relief valve 122 also has a spool 122k having three lands 122h, 122i and 122j slidably received by the bore sections 122e, 122f and 122g, respectively, and a spring 122m biasing the spool 122k. If desired, the port 122c may be connected via a branch 136 to the lock-up duty solenoid 128. In this embodiment, however, the branch 136 is blocked by a separator 136s.

In addition to the inlet port 124a, the torque converter relief valve 124 has an outlet port 124b, a spool 124d with a single land 124c and a spring 124 moving the spool 124d in a direction to cover the outlet port 124b. The outlet port 124b is connected via a lubricant relief ball valve 144 to the suction side of the pump 101. Oil discharged from the outlet port 12b is supplied, as lubricant, to a system such as power train and V-belt, and differential.

In addition to the inlet port 128a, the lock-up duty solenoid valve 128 has a drain port 128c and an outlet port

128b connected to a port 150a of the shift command valve 150. When duty (DUTY₁₂₈) of the solenoid valve 128 is 100%, a needle closes the drain port 128c, allowing the full supply of the pilot pressure from the inlet port 128a to the outlet port 128b. When the duty DUTY₁₂₈ is 0%, the needle opens the drainport 128c, draining oil from the outlet port 128b. The duty DUTY₁₂₈ can be adjusted to any value falling in a range from 0% to 100%. Pressure appearing at the outlet port 128b, which may be called as a lock-up control pressure, varies continuously between the lowest level and the highest level as high as the pilot pressure as DUTY₁₂₈ varies continuously between 0% and 100%. In other words, the lock-up control pressure is proportional to DUTY₁₂₈.

The lock-up control valve 126 has a bore 126a and a reduced diameter bore 126h. The bore 126a is formed with an inlet port 126b, a first outlet port 126c, a second outlet port 126c, a third outlet port 126d, a fourth outlet port 126e, a fifth outlet port 126f, and a drain port 126g. The reduced diameter bore 126h is formed with a port 126i connected via an orifice 148 to the outlet port 126c and with a port 126j connected to an outlet port 150b of the shift command valve 150, which is in turn connected to the inlet port 150a. The inlet port 126b is connected to the outlet port 122d of the clutch release valve 122. The outlet port 126c is connected to a lock-up chamber 12a. The outlet port 126d is connected to the fluid coupling 12. The outlet port 126e is connected to a cooler 146. The outlet port 126f is connected to the lubrication system. The lock-up control valve 126 has a spool 126s having four lands 126m, 126n, 126o and 126p received in the bore section 126a and a land 126r received in the reduced diameter bore section 126h. The lock-up control valve 126 also has a spring 126t biasing the spool 126s. A relief valve 152 is connected to a passage 149 connected to the outlet port 126d at one end and to the fluid coupling 12 at the other end.

When the duty DUTY₁₂₈ is 0%, the lock-up control pressure is zero or almost zero, allowing the spring 126t to move the spool 126s to a spring set position as illustrated in FIG. 2B. In this position, pressurized oil supplied from the outlet port 122d of the clutch relief valve 122 to the inlet port 122b of the lock-up control valve 126 is supplied via the outlet port 126c to the lock-up chamber 12a. This results in disengagement of the lock-up clutch 12d. As the duty DUTY₁₂₈ increases, the spool 126s moves against the spring 126t, increasing fluid communication between the outlet port 126c and the drain port 126g, resulting in gradual decrease in pressure within the lock-up chamber 12a. When the duty DUTY₁₂₈ is 100%, the lock-up control pressure is as high as the pilot pressure, positioning the spool 126s at a lock-up position against the bias of the spring 126t. In the lock-up position, the lock-up chamber 12a is drained, resulting in engagement of the lock-up clutch 12d.

In addition to the inlet port 104a connected to the line pressure regulator valve 112, the manual selector valve 104 has a R range port 104b, a D range port 104c, a L range port 104d and two axially remotest drainages 104e and 104f. Cooperating with these ports is a manually operable spool 104i having two lands 104g and 104h. The spool 104i moves in response to positioning a selector lever (not shown) to any one of six detent position, namely P (Park), R (Reverse), N (Neutral), D (Drive), 2 and L. The R range port 104b is connected via a reverse brake control valve 140 to the reverse brake 50. The D and L range ports are connected via a forward clutch control valve 142 to the forward clutch 40.

In addition to the inlet port 129a, the clutch duty solenoid 129 has a drain port 129c and an outlet port 129b connected

to ports 140n and 142h of the reverse brake control valve 140 and forward clutch control valve 142, respectively. When duty (DUTY₁₂₉) of the solenoid valve 129 is 100%, a needle closes the drain port 129c, allowing the full supply of the pilot pressure from the inlet port 129a to the outlet port 129b. When the duty DUTY₁₂₉ is 0%, the needle opens the drain port 129c, draining oil from the outlet port 129b. The duty DUTY₁₂₉ can be adjusted to any value falling in a range from 0% to 100%. Pressure appearing at the outlet port 129b, which may be called a clutch control pressure, varies continuously between the lowest level and the highest level as high as the pilot pressure as DUTY₁₂₉ varies continuously between 0% and 100%. In other words, the clutch control pressure is proportional to DUTY₁₂₉.

In addition to the port 140h, the reverse brake control valve 140 has an inlet port 140a connected to the R range port 140b, an outlet port 140d and a feedback port 140g connected via an orifice 140f to the outlet port 140d. The outlet port 140d is connected to the reverse brake 50. The reverse brake control valve 140 has a spool 140m having three lands 140i, 140j and 140k, and a spring 140n biasing the spool 140m. Fluidly disposed between the outlet port 140d and the reverse brake 50 are a discharge flow restricting orifice 140b bypassed by a one-way check valve 140o and a supply flow restricting orifice 140c bypassed by a one-way check valve 140p. Oil to be supplied to the reverse brake 50 passes through the check valve 140o and supply flow restricting orifice 140c. Oil to be discharged from the reverse brake 50 passes through the check valve 140p and discharge flow restricting orifice 140b.

In addition to the port 140h, the forward clutch control valve 142 has a port 140b connected to the D range port 140c, a port 140e connected to the L range port 140d, an outlet port 140d connected to the forward clutch 40, and a feedback port 140g connected via an orifice 140f to the outlet port 140d. The forward clutch control valve 142 has a spool 140m having three lands 140i, 140j and 140k, and a spring 140n biasing the spool 140m. Fluidly disposed between the outlet port 140d and the forward clutch 40 are a discharge flow restricting orifice 142b bypassed by a one-way check valve 142o and a supply flow restricting orifice 142c bypassed by a one-way check valve 142p. Oil to be supplied to the forward clutch 40 passes through the check valve 140o and supply flow restricting orifice 142c. Oil to be discharged from the forward clutch 40 passes through the check valve 142p and discharge flow restricting orifice 142b. It is to be noted that, at the shift position D, the L range port 142d is drained, and the port 142e becomes a drain port and the port 142b becomes an inlet port. At the shift position L, the D range port 142c is drained, and the port 142b becomes a drain port and the port 142e becomes an inlet port.

The reverse brake and forward clutch valves 140 and 142 are identical in construction and function. Let us now consider the case when the shift position D is chosen. The D range port 104c is supplied with line pressure from the inlet port 104a and the L range port 104d is drained. If the clutch control pressure at the port 142h is zero (DUTY₁₂₉=0%), the forward clutch control valve 142 generates at the outlet port 142d a servo activating pressure that is determined by a differential area between the lands 142i and 142j and the bias force of the spring 142n. As the clutch control pressure increases, the force acting on the land 142i against the spring 142n increase and thus the servo activating pressure at the port 142d decreases. When the clutch control pressure is as high as the pilot pressure (DUTY₁₂₉=100%), the servo acting pressure is zero or almost zero. From the preceding,

it is now understood that the servo activating pressure supplied to the forward clutch **40** or reverse brake **50** can be altered by varying the duty DUTY₁₂₉.

The stepping motor **108** and the three duty solenoids **120**, **128** and **128** are under the control of a CVT controller **300**.

Referring to FIG. 3, the controller **300** receives output signals of a plurality of sensors and switch which detect operating variables which the motor vehicle is involved in. The plurality of sensors and switch comprise an accelerator (ACC) position sensor **401**, an engine speed sensor **301**, a vehicle speed sensor **302**, a throttle position sensor **303**, a shift position switch **304**, a turbine speed sensor **305**, an engine coolant temperature sensor **306**, a brake sensor **307**, and an angular position sensor in the form of a rotary encoder **318**.

The ACC position sensor **401** detects an angle through which the gas pedal **2** is depressed and generates, as operator power request command, an output signal indicative of the detected position or angle. The engine speed sensor **301** detects an engine revolution speed by counting the number of engine ignition spark pulses. The vehicle speed sensor **302** detects a vehicle speed by measuring rotational speed of the output shaft of the continuously variable transmission. The throttle opening sensor **303** detects an opening degree of the throttle **4**. The shift position switch **304** detects which of different shift positions P, R, N, D, 2 and L the manual selector valve **104** is placed at. The turbine speed sensor **305** detects a revolution speed of the turbine shaft **13**. The engine coolant temperature sensor **306** generates an output when the engine coolant temperature is lower than a predetermined value. The brake sensor **307** detects whether a foot brake is pressed or not. The rotary encoder **318** detects an angular position of the stepping motor **108** and generates an output representative of the detected angular position.

Outputs of the ACC position sensor **401** and throttle position sensor **303** are converted into a digital form at A/D converters **402** and **310**, respectively, and fed to an input interface **311**. Outputs of the engine speed sensor **301**, vehicle speed sensor **302**, and turbine speed sensor **305** are supplied via the associated wave shapers **308** and **309** to the input interface **311**. Outputs of the shift position switch **304**, engine coolant temperature sensor **306** and brake sensor **307** are supplied to the input interface **311**.

The controller **300** is of a microcomputer based control unit and comprises the input interface **311**, a CPU (central processor unit) **313**, a reference pulse generator **312**, a ROM (random access memory) **315**, and an output interface **316**. These elements are interconnected by an address bus **319** and a data bus **320**. The reference pulse generator **312** generates reference pulse on which the CPU **313** operates. The RAM **317** temporarily stores information from various sensors and switches and parameters necessary for control. Outputs of the controller **300** are supplied via the output interface **316** to a stepping motor driver **317** and also to the pressure modifier duty solenoid valve **120**, lock-up duty solenoid valve **128** and clutch duty solenoid valve **129**.

FIG. 4 shows variation of the CVT ratio C_P against variation of an actual step number P_A that is determined by the CVT controller **300** based on an angular position I_S of the stepping motor **108** detected by the position sensor or rotary encoder **318**. The setting is such that the CVT ratio C_P is the largest or maximum ratio C_{MAX} when the actual step number P_A is zero and decreases toward the smallest or minimum ratio C_{MIN} as the actual step number P_A increases from zero.

The CVT controller **300** determines a desired speed SP of movement of the stepping motor, which may be called as a

step rate, and generates a train of clockwise (CW) or counterclockwise (CCW) pulses whose pulse rate is variable in response to the determined desired speed SP. The train of pulses is supplied to the stepping motor driver **317**, causing the stepping motor **108** to move at the desired speed SP determined by the CVT controller **300**. The direction of movement of the stepping motor **108** is determined by the CVT controller **300** as a result of comparison of actual step number P_A with a desired step number P_D , and an appropriate one of CW and CCW pulses is chosen. The amount of movement of the stepping motor **108** is determined by the CVT controller **300** as a result of computation of a step number error P_E expressed in terms of number of steps, i.e., $|P_D - P_A|$.

The CVT controller **300** controls the speed of movement of the stepping motor **108**, controlling the speed of movement of the rod **182**, thus controlling the ratio rate of the CVT (see FIG. 10). The faster the stepping motor **108** moves the rod **182**, the faster the ratio C_P of the CVT changes.

The CVT controller **300** sets the desired speed SP equal to a maximum or highest speed SP_{MAX} for quick response to a shift request command. However, the desired speed SP is set equal to a speed, including zero, lower than the maximum speed SP_{MAX} in response to a quick release of the gas pedal **2** for improved engine brake performance. The higher the rate of change per unit time of the position of the gas pedal **2** toward the released position thereof is, the lower the desired speed SP is. The power request command may be determined by detecting depression degree or position of the gas pedal **2** and the rate at which the power request command changes may be easily determined by computing a change in the depression degree per unit time in the CVT controller **300**.

A preferred implementation of the present invention may be understood with reference to a main control routine illustrated the flow diagram of FIGS. 5A and 5B and a sub control routine illustrated by the flow diagram of FIG. 12.

At input box **502**, the controller **300** inputs information of transmission shift position, namely, P (Park), R(Reverse), N(Neutral), D(Drive), 2 and L, after performing reading operation of output of the shift position switch **314** through the input interface **311**. The controller **300** next determines, at box **504**, whether the operator has chosen one of D, 2, L and R or one of P and N.

If the operator has chosen P or N, the duty DUTY₁₂₈ to be output to the lock-up duty solenoid **128** is set equal to 0% at box **506**. The controller **300** then compares the actual step number P_A with desired step number P_D at box **630**.

If the operator has chosen D or 2 or L or R, the controller **300** inputs information of throttle opening degree TH at box **508**, vehicle speed V (or output shaft speed) at box **510**, and stepping motor angular position I_S at box **511a**. At box **508**, reading operation of output of the throttle opening degree or position sensor **303** is performed through the input interface **311**. At box **511a**, reading operation of output of the angular position sensor **318** is performed through the input interface **311**.

At box **511b**, the controller **300** computes actual step number P_A which may be expressed as a function of, as a variable, stepping motor angular position I_S . The actual step number P_A can take 0 (zero) or a range of consecutive position integers from 1. The stepping motor **108** can take various angular positions in discrete manner. The values which may be taken by the actual step number P_A are assigned to all of angular positions which may be taken by the stepping motor **108**, respectively. Thus, using this pre-

determined relationship, the actual step number P_A is determined by the angular position I_S .

A box 512, the controller 300 inputs information of engine speed N_E . In box 512, reading operation of output of the engine speed sensor 301 is performed through the input interface 311.

The controller 300 computes the CVT ratio C_P at box 513a, engine torque T_E at box 513a, and line pressure P_L at box 513c. In box 513a, the controller 300 performs a table look-up operation of FIG. 4 using the actual step number P_A to determine CVT ratio C_P . In box 513b, the controller 300 performs a table look-up operation of FIG. 6 using throttle opening degree TH and engine speed N_E to determining engine torque T_E . In box 513c, the controller 300 performs a table look-up operation of FIG. 7 using engine torque T_E and CVT ratio C_P and determines duty DUTY₁₂₀ to be output to the pressure modifier duty solenoid valve 120 for generation of line pressure P_L required.

At box 514, the controller 300 inputs information of turbine speed N_t . In box 514, reading operation of output of the turbine speed sensor 305 is performed through the input interface 311.

The controller 300 computes a deviation N_D at box 516 by subtracting the turbine speed N_t from the engine speed N_E . At box 518, a table look-up operation of FIG. 8 is performed using the throttle opening degree TH and the vehicle speed V to determine a lock-up on vehicle speed V_{ON} and a lock-up off vehicle speed V_{OFF} . The controller 300, next, determines, at box 520, whether a lock-up flag LUF is set equal to 1.

If the lock-up flag LUF is set equal to 1, the controller 300 determines, at box 544, whether the vehicle speed V is less than the lock-up off vehicle speed V_{OFF} . If not, the controller 300 determines, at box 522, whether the vehicle speed V is greater than the lock-up on vehicle speed V_{ON} .

If, at box 544, the vehicle speed V is less than the lock-up off vehicle speed V_{OFF} ($V < V_{OFF}$), the duty DUTY₁₂₈ to be output to the lock-up duty solenoid valve 128 is set equal to 0% at box 540. The controller 300 clears or resets the lock-up flag LUF at box 542. The controller 300, next, goes to box 601. If, at box 544, the vehicle speed V is not less than the lock-up off vehicle speed V_{OFF} , the duty DUTY₁₂₈ to be output to the lock-up duty solenoid valve 128 is set equal to 100% at box 546. The controller 300, next, goes to box 601.

If, at box 522, the vehicle speed V is greater than the lock-up on vehicle speed V_{ON} , the controller 300 goes to box 524. If, at box 522, the vehicle speed V is not greater than the lock-up on vehicle speed V_{ON} , the controller 300 goes to box 540.

In box 524, the controller 300 computes a desired deviation e by subtracting a preset first target value N_{m1} from the deviation N_D obtained at box 516. At box 526, the controller 300 performed a table look-up operation of a first feedback gain map to determine a first feedback gain G_1 for the desired deviation e . The controller 300, next, determines at box 528 whether the deviation N_D is less than a preset threshold value N_0 . If, at box 528, the deviation N_D is less than the preset threshold value N_0 , the controller 300 goes to box 530. If, at box 528, the deviation N_D is not less than the preset threshold value N_0 , the controller 300 goes to box 538.

In box 530, the duty DUTY₁₂₈ to be output to the lock-up duty solenoid valve 128 is increased by a preset small value α (alpha). The controller 300, next, sets the duty DUTY₁₂₈ equal to 100% at box 534 and sets the lock-up flag LUF equal to 1 at box 536. Next, the controller 300 goes to box 601.

In box 538, the duty DUTY₁₂₈ is decreased by the preset small value α (alpha). The controller 300, next, goes to box 601.

In box 601, the controller 300 determines whether anti-skid operation is in progress. The vehicle is installed with a brake system which is able to perform anti-skid operation to control slip rate of road wheels. The controller 300 communicates with a controller of the brake system to determine whether anti-skid operation is in progress.

If anti-skid operation is not in progress, the controller 300 determines at box 602 whether vehicle speed V is less than a threshold vehicle speed V_0 of ratio control. This threshold vehicle speed V_0 ranges from 2 km/h to 3 km/h and thus lower than lock-up on and lock-up off vehicle speeds V_{ON} and V_{OFF} .

If anti-skid operation is in progress, the duty DUTY₁₂₉ to be output to the clutch duty solenoid valve 129 is set equal to 100% at box 601a. The controller 300, next, goes to box 602.

If, at box 602, vehicle speed V is less than threshold vehicle speed V_0 , the controller 300 determines that creep control is required and goes to box 604. If vehicle speed V is not less than threshold vehicle speed V_0 , the controller 300 determines that ratio control is not required and goes to box 624.

In box 604, the controller 300 determines whether throttle opening degree TH is less than a critical throttle opening degree TH_0 of idle operation (see FIG. 9). If throttle opening degree TH is less than the critical throttle opening degree TH_0 , the controller 300 goes to box 610. If throttle opening degree TH is not less than the critical throttle opening degree TH_0 , the controller 300 sets the duty DUTY₁₂₉ equal to 0% at box 606. The controller 300, next, sets desired step number P_D equal to 0 (zero) at box 608. Then, the controller 300 goes to box 630.

In box 610, the controller 300 determines whether actual step number P_A is 0 (zero). If actual step number P_A is zero ($P_A=0$), the controller 300 goes to box 612. If actual step number P_A is not equal to zero, the controller 300 sets a desired step motor speed SP equal to maximum or highest speed SP_{MAX} at box 619. The desired step motor speed SP is expressed by a change in step number per unit time.

In box 612, the controller 300 computes desired deviation e by subtracting a preset second target value N_{m2} from deviation N_D . Next, the controller 300 performs a table look-up operation of a second feedback gain map to determine a second feedback gain G_2 for the desired deviation e . The controller 300, next, computes at box 616 a predetermined equation containing as variables the desired deviation e and second feedback gain G_2 and sets the duty DUTY₁₂₉ equal to the result of this computation. Then, the controller 300 goes to box 638.

In box 638, the controller 300 outputs DUTY₁₂₀, DUTY₁₂₈ and DUTY₁₂₉ to pressure modifier duty solenoid 120, lock-up duty solenoid 128 and clutch duty solenoid 129, respectively.

Varying the duty of clutch duty solenoid 129 alters degree of engagement of the forward clutch 40 if it is selected by the manual selector valve 104 or the reverse brake 50 if it is selected by the manual selector valve 104. Complete engagement is accomplished if DUTY₁₂₉ is 0%, while complete disengagement if DUTY₁₂₉ is 100%. The degree of slip of forward clutch 40 or reverse brake 50 increases continuously as DUTY₁₂₉ increases so that the slip can be adjusted to a desired degree corresponding to the value of DUTY₁₂₉.

Creep control is initiated if $P_A=0$ (at box 610) under conditions when $TH < TH_0$ (at box 604) with $V < V_0$ (at box 602) and the degree of slip is determined at box 616.

In box 624, the controller 300 determines whether shift position D is chosen. If shift position D is chosen, the controller 300 determines, at box 626, a desired step number P_D and a desired motor speed SP by executing a sub-routine shown in FIG. 12 wherein the desired step number P_D is determined by performing a table look-up operation of D-range shift pattern against throttle opening degree TH and vehicle speed V. The controller 300, next, goes to box 630.

If, at box 624, shift position D is not chosen, the controller 300 determines whether shift position 2 is chosen at box 625. If shift position 2 is chosen, the controller 300 determines, at box 627, a desired step number P_D by executing a sub-routine substantially the same as that shown in FIG. 12 wherein the desired step number P_D is determined by performing a table look-up operation of 2-range shift pattern against throttle opening degree TH and vehicle speed V. The controller 300, next, goes to box 630.

If, at box 625, shift position 2 is not chosen, the controller 300 determines whether shift position L is chosen at box 639. If shift position L is chosen, the controller 300 determines, at box 628, a desired step number P_D by performing a table look-up operation of L-range shift pattern against throttle opening degree TH and vehicle speed V and sets a desired motor speed SP equal to SP_{MAX} . The controller 300, next, goes to box 630.

If, at box 639, shift position L is not chosen, the controller 300 determines, at box 640, a desired step number P_D by performing a table look-up operation of R-range shift pattern against throttle opening degree TH and vehicle speed V and sets desired motor speed SP equal to SP_{MAX} . The controller 300, next, goes to box 630.

The D-range, 2-range, and L-range shift patterns are explained in connection with FIG. 9.

In FIG. 9, the vertical axis represents engine speed N_E and the horizontal axis represents vehicle speed V. The fully drawn line C_{Hi} illustrates a linear relation between engine speed N_E and vehicle speed V when CVT ratio C_P is fixed to C_{Hi} ($=C_{MAX}$). The fully drawn line C_{DLO} illustrates another linear relation between engine speed N_E and vehicle speed V when CVT ratio is fixed to C_{DLO} ($=C_{MIN}$). The fully drawn line C_{2LO} illustrates still another linear relation between engine speed N_E and vehicle speed V when CVT ratio is fixed to C_{2LO} that is smaller than C_{DLO} . Throttle opening degree TH is used as a parameter and an arrow TH indicates a direction of increase of this parameter. The throttle opening degree TH takes the critical throttle opening degree TH_0 when the vehicle speed is at the threshold vehicle speed v_0 with the CVT ratio at C_{Hi} . On the horizontal axis, the threshold vehicle speed V_0 is illustrated. Also illustrated on the horizontal axis are a vehicle speed V_{D1} at which CVT ratio C_{DLO} is maintained at critical throttle opening degree TH_0 , and a vehicle speed V_{21} at which CVT ratio C_{2LO} is maintained at critical throttle opening degree TH_0 .

The D-range shift pattern is a map containing a set of values which CVT ratio may take against various combinations of throttle opening degree TH and vehicle speed V. This set of values can be plotted in FIG. 9 within a zone between the fully drawn lines C_{Hi} and C_{DLO} . A curve C_{PTN} drawn by the two-dot chain line illustrates gradual increase of engine speed N_E versus vehicle speed V during gradual acceleration with the shift position D chosen by the operator.

The 2-range shift pattern is a map containing another set of values which CVT ratio may take against various com-

binations of throttle opening degree TH and vehicle speed V. This another set of values can be plotted in FIG. 9 within a zone between the fully drawn lines C_{Hi} and C_{2LO} .

The L-range shift pattern is a map containing still another set of values which CVT ratio may take against various combinations of throttle opening degree TH and vehicle speed V. This still another set of values can be plotted in FIG. 9 on the fully drawn line C_{Hi} . In other words, CVT ratio is held at C_{Hi} according to L-range shift pattern.

The D-range, 2-range and R-range shift patterns are programmed in the form of map in ROM 314 of the controller 300.

In box 630, the controller 300 compares actual step number P_A with the desired step number P_D . If actual step number P_A is equal to the desired step number P_D , the controller 300 outputs DUTY₁₂₀, DUTY₁₂₈ and DUTY₁₂₉. If actual step number P_A is larger than the desired step number P_D and a downshift is required, the controller 300 goes to box 620. If actual step number P_A is smaller than desired step number P_D and an upshift is required, the controller 300 goes to box 632.

In box 632, the controller 300 forms a ratio rate command to move stepping motor 108 in upshift direction. The ratio rate command contains information of step number error or deviation expressed by $P_D - P_A$, desired motor speed or step rate SP, and direction of movement of stepping motor 108. The controller 300 next, goes to box 636. In box 636, the controller 300 initiates execution of a stepping motor control routine under conditions set by the ratio rate command and repeats execution thereof to output pulse at such a pulse rate as to achieve the step rate until actual step number P_A becomes equal to the desired step number P_D . The controller 300, next, goes to box 638.

If, at box 630, actual step number P_A is larger than desired step number P_D , the controller 300 goes to box 620.

In box 620, the controller 300 forms a ratio rate command to move stepping motor 108 in downshift direction. The controller 300, next, initiates, at box 636, execution of stepping motor control routine under conditions set by the ratio rate command and repeats execution thereof to output pulse at such a pulse rate as to achieve the step rate until actual step number P_A becomes equal to desired step number P_D . The controller 300, next, goes to box 638.

FIG. 11 illustrates the characteristic of coefficient K. In FIG. 11, the vertical axis represent the coefficient K. The horizontal axis represents a throttle rate H, i.e., a change in throttle opening degree TH per unit time. This characteristic of coefficient K is programmed in the form of a map in ROM 314 of the controller 300.

The sub-routine illustrated by the flow diagram in FIG. 12 is executed at box 626 if the shift position D is chosen. The operator depresses or releases the gas pedal 2 sending a power request command to the controller 300 through the accelerator position sensor 401. In FIG. 12, at input box S1, the controller 300 inputs a new data A(n) of the power request command and an old data A(n-1) of the power request command. In this embodiment, the power request command is sensed by the accelerator position sensor 401. Output of the accelerator position sensor 401 represents position of the gas pedal 2. In box S1, the controller 300 performs reading operation of the output of the accelerator position sensor 401 through the input interface 311 to input the new data A(n).

In box S2, the controller 300 computes a power request command rate H, i.e., a change in power request command per unit time, expressed by the following equation:

$$H=(A(n-1)-A(n))/\text{delta } T$$

where: delta T represents a time interval between repetition of execution of the main routine illustrated by the flow diagram of FIGS. 5A and 5B.

Alternatively, the power request command rate H may be determined from output of an accelerometer which measures the degree of acceleration which the gas pedal 2 is subjected to.

The controller 300, next, determines coefficient K, at box S3, by performing a table look-up operation of FIG. 11 using the throttle rate H as a variable. From FIG. 11, it is seen that the coefficient K is equal to 1 (one) and unaltered against variation of the throttle rate H as long as it is greater than a preset value H_0 that is less than 0 (zero). If the throttle rate H is less than the present value H_0 , the coefficient K is less than 1 (one). The less the throttle rate H is, the less the coefficient K is. In other words, the coefficient K decreases as the speed at which the operator releases the gas pedal 2 increases.

In box S4, the controller 300 computes a desired stepping motor speed SP which is expressed as a product of the coefficient K and the maximum speed SP_{MAX} . The desired motor speed SP is expressed in terms of a pulse number per unit time. The controller 300, next, determines a stepping number error P_E at box S5. In this case, since the shift position D is chosen, the D-range shift pattern is accessed to perform a table look-up operation to determine desired CVT ratio C_D for the throttle opening degree TH obtained at block 508 and vehicle speed V obtained at block 510.

In box S5, the controller 300 inputs the already determined desired step number P_D and the actual step number P_A determined at box 511b, and calculates a deviation $P_E (=P_D - P_A)$.

In box S6, the controller 300 determines whether the absolute value of step number error $|P_E|$, i.e., $|P_D - P_A|$, is less than or equal to the desired speed SP or not. If this is the case, the desired speed SP is set equal to the absolute value of deviation $|P_E| (=|P_D - P_A|)$. In other words, the desired speed SP is decreased. The controller 300, next, updates the old data $A(n-1)$ of power request command with the new

data $A(n)$ at box S8. If, at box S6, the absolute value of step number error $|P_D - P_A|$ is greater than the desired speed SP, the controller 300, directly goes to box S8. The controller 300 next, returns to the main routine at box 630 in FIG. 5B.

In box 627 of FIG. 5B, the controller 300 executes a sub-routine substantially the same as the sub-routine illustrated by the flow diagram in FIG. 12. In this case, the 2-range shift pattern is used in stead of the D-range shift pattern in determining a desired step number P_D .

In FIG. 13, let us now consider a moment upon release of the gas pedal 2 during forward travel with the shift position D chosen. At this moment, the throttle opening degree TH decreases and the output shaft torque T_O drops correspondingly. Immediately after this moment, the controller 300 determines a desired speed SP at box 626 and turns the stepping motor 108 at the desired speed SP to change the CVT ratio C_P in upshift direction at the rate determined by the speed SP. The rate of change per unit time of CVT ratio is suppressed and thus the output shaft torque T_O increases at a gradual rate without inducing any undesired peak. In FIG. 14, such undesired peak of output shaft torque T_O appears immediately after the CVT ratio changes at unsuppressed ratio rate.

What is claimed is:

1. A control system for a vehicle drive-train with an engine operable with a gas pedal and a continuously variable transmission (CVT) comprising:

a controller means for determining a rate of change per unit time of the positions of the gas pedal, and

a ratio control element positionable to various command positions in response to the rate of change per unit time determined by the controller means to establish various CVT ratios,

wherein the controller means decreases the rate of change per unit time of the CVT ratio as the rate of change per unit time of the positions of the gas pedal toward a released position thereof increases.

2. A control system as claimed in claim 1, wherein the controller means limits the rate of change per unit time of CVT ratio.

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